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IGCC PERFORMANCE COMPARISON FOR VARIATIONS IN GASIFIER TYPE AND GAS TURBINE FIRING TEMPERATURE

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SUMMARY

Performance estimates were made for a series of integrated coal gasification combined-cycle (IGCC) power systems using three generic types of coal gasification subsystems.

The objectives of this study were (1) to provide a self-consistent comparison of IGCC systems using different types of gasifiers and different oxidants and (2) to use this framework of cases to evaluate the effect of a gas-turbine firing temperature and cooling approach on overall system efficiency.

The basic IGCC systems considered included the use of both air- and oxygen-blown versions of a fluidized bed gasifier, represented by the Westinghouse design, and an entrained-bed gasifier, represented by the Texaco design. Also considered were systems using an oxygen-blown, fixed-bed gasifier, represented by the British Gas Corporation (BGC) slagging gasifier. All of these gasifiers were integrated with a combined cycle using a gas-turbine firing temperature of 1700 K (2600° F) and a compressor pressure ratio of 16:1. Steam-turbine throttle conditions were chosen to be 16.6 MPa/811 K (2400 psia/1000° F) with a single reheat to 810 K (1000° F).

Some of these cases were modified to allow the evaluation of the effect of gas-turbine firing temperature. Turbine firing temperatures from state-of-the-art 1365 K (2000° F) to an advanced-technology 1920 K (3000° F) were analyzed. A turbine-cooling technology that maintains metal temperatures below acceptable limits was assumed for each level of firing temperature. System performance comparisons were made using three advanced turbine-cooling technologies for the 1920 K (3000° F) firing temperature.

The results indicate that the IGCC using the BGC gasifier had the highest net system effiency (42.1 percent) of the five gasification cases considered. The other four cases had net system efficiencies in the 40.0-percent range.

In all cases net system efficiency increased as firing temperatures increased. The increase is greatest for the fixed-bed gasifier because of the higher chemical energy content of its fuel gas. The air-blown version of the fluid-bed gasifier has an efficiency advantage over its oxygen-blown counterpart, but the advantage decreases as the firing temperature increases.

The use of advanced convective cooling technology, at a 1920 K (3000° F) firing temperature, results in higher system performance than either water cooling in the fixed-bed system or transpiration cooling in the fluid-bed gasifier system.

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Sensitivity parameters developed from the cases using the 1700 K (2600° F) gas-turbine firing temperature indicate that it is particularly important to improve the individual performance of both the gas turbine and steam turbine subsystems. The gas-turbine system performance can be improved by increasing firing temperature in conjunction with advanced cooling techniques. The steam-turbine system can be improved by careful selection of throttle and reheat conditions that can use the high-quality heat available from the raw gas cooler and the heat-recovery steam generator.

INTRODUCTION

Over the past several years, numerous analyses of integrated coal gasification, combined-gas-turbine-steam-turbine power systems (IGCC systems) have been reported in the literature (e.g., refs. 1 to 5). These studies indicate that IGCC systems can use a wide variety of coal, including high-sulfur coal, in an environmentally attractive manner and compete favorably with direct coal-fired boiler/steam-turbine plants using stack gas desulfurization. These IGCC systems analyses have considered the use of several generic types of coal gasification systems. Among them are fixed-bed, fluidized-bed, and entrained-bed gasifiers using either air, oxygen, or a mixture of air and oxygen (air enriched) as the oxidant. The integration of these gasifiers with combined cycles is complex because of the many alternative configurations and their relative effects on performance and cost.

Also, the differences in other assumptions made in the various analyses cause the results to vary widely. For example, overall system efficiencies from 30 to 40 percent have been reported for IGCC systems using a 1365 K (2000° F) gas-turbine firing temperature. These differences are attributable not only to the use of different types of gasifiers, but also to differences in overall system configuration and subsystem design parameters. Direct evaluation of the effects of a specific assumption or design parameter is difficult because the above analysis generally involves changes in more than one parameter or assumption.

The objectives of the present study were (1) to provide a consistent comparison of IGCC systems using different gasifer types and different oxidants and (2) to use this framework of IGCC cases to evaluate the effects of gas-turbine firing temperatures and cooling approaches on overall system efficiency.

The basic IGCC systems considered include the use of both air and oxygen-blown versions of a fluidized-bed gasifier, represented by the Westinghouse gasifier design, and an entrained-bed gasifier, represented by the Texaco gasifier. Also considered were systems using an oxygen-blown fixed-bed gasifier represented by the British Gas Corporation (BGC) slagging gasifier. All of these gasifiers were integrated with a combined cycle using a gas-turbine firing temperature (combustor-exit temperature) of 1700 K (2600° F) and a compressor pressure ratio of 16:1. Steam-turbine throttle pressure and temperature were chosen to be 16.6 MPa (2400 psia) and 810 K (1000° F), respectively, with a single reheat to 810 K (1000° F).

Some of these IGCC cases were then modified to evaluate the effects of gas-turbine firing temperature. For this purpose both the air- and oxygen-

blown Westinghouse gasifiers and the BGC slagging gasifier were considered. Turbine firing temperatures from the state-of-the-art 1365 K (2000° F) to an advanced-technology 1920 K (3000° F) were analyzed. A turbine cooling technology that maintains metal temperatures below acceptable limits was assumed for each level of firing temperature. System performance comparisons were made using three advanced turbine-cooling technologies for the 1920 K (3000° F) firing temperature.

Sensitivity parameters were developed from a simplified analytical model which relates the uncoupled effect on overall system performance of changes in some of the major subsystem parameters. The purpose of this analysis was to identify the subsystem parameters which, if improved, would have the greatest effect on overall system performance.

The operating parameters and raw gas composition for each of the gasifier configurations were obtained from the literature (refs. 1, 3, and 7). The gas-turbine performance was obtained from NASA Lewis performance code. The steam-turbine performance was obtained from the Presto computer code described in reference 6. Other key performance data, such as subsystem auxiliary power requirements, were taken from references 1, 4, 5, and 7 to 11.

ASSUMPTIONS AND CONSTRAINTS

The following assumptions were used in developing the configuration of each IGCC system reported herein:

1. Each IGCC system was configured to be consistent with the particular gasifier heat and mass balance and the composition data available in the literature. Within the constraint of the availability of data, every effort was made to configure each system in a consistent manner.

2. The fuel gas temperature to the gas turbine combustor was 589 K (600° F).

3. Ninety-five percent of the hydrogen sulfide (H_2S) and 100 percent of the ammonia (NH_3) contained in the raw fuel gas were removed in the cleanup system. The 95-percent H_2S removal exceeds the new source performance standards (NSPS) for coal-fired plants and is well within the reported capability of several cleanup systems.

4. Steam-turbine throttle conditions were 10.0 MPa and 783 K (1450 psia and 950° F) with a single reheat to 785 K (950° F) for the 1365 K (2000° F) gas-turbine firing temperature. Steam-turbine throttle conditions of 16.6 MPa and 811 K/ 810 K (2400 psia and 1000° F/1000° F) were used for 1700 K (2600° F) and 1920 K (3000° F) firing temperatures.

5. The factors used in estimating the various subsystem auxiliary power requirements are listed in table I along with the source of information for these factors.

6. The minimum temperature difference in all heat exchangers was 28 K (50.0° F).

7. A steam-condenser pressure of 0.008 MPa (1.16 psia) was used in all cases.

8. The gas-turbine and compressor polytropic efficiency was 89 percent for all cases except those involving transpiration cooling. For the transpiration-cooled case, each 1 percent of coolant flow to a particular row decreased the polytropic efficiency for each vane and blade row, respectively, by 1.5 percent.

9. The minimum raw-fuel-gas temperature exiting the raw gas cooler was 450 K (350° F).

GENERAL DESCRIPTION OF CASES

Fourteen powerplant configurations were analyzed. These configurations included selected combinations of the five gasifiers, three gas-turbine firing temperatures, and five turbine-cooling methods. The basic operating parameters for each of these configurations are shown in table II. The code used to identify each configuration is defined in table III.

Schematics of IGCC systems using the five gasifiers at a 1700 K (2600° F) firing temperature are shown in figures 1 to 5. Except for variations in combined-cycle operating parameters, these schematics are representative of all cases considered. Each case consists of a coal gasifier and its oxidant supply system, a raw gas cooler (except for the BGC slagging gasifier case), a gas cleanup system, and a combined cycle. The combined cycle consists of an advanced high-temperature gas turbine, a heat-recovery steam generator (HRSG), and a reheat steam turbine.

Each IGCC system configurations shown in figures 1 to 5 is somewhat different. An objective of this analysis was to limit these differences in configurations to those which were necessitated by variations in gasifier characteristics (e.g., gasifier exit temperatures). In some cases, however, configuration differences were introduced by limitations in the availability of gasifier data in the literature. For example, the gasifier oxidant inlet temperature used was that for which gasifier heat and mass balances were available. In practice, this inlet temperature would be chosen to yield the best interface between the gasifier, the oxidant delivery system, and the combined cycle. The further heating of the gasifier oxidant to 770 K (925° F) in the HRSG shown in figure 2 was done not because of an effort to improve system performance, but because the gasifier data used were based on an oxidant inlet temperature of 770 K (925° F). Within these constraints every effort was made to configure and compare the systems in a consistent manner.

All cases were sized using a coal input flow rate of 160 500 kg/hr (353 900 lb/hr). This coal flow rate, for the Texaco gasifiers, yields a net electrical output of approximately 500 MW, which represents the average-size coal-fired-base-load unit ordered by utilities in recent years.

The gasifier oxidant supply for the air-blown cases is obtained by cooling bleed air from the gas-turbine compressor discharge, compressing it with a motor-driven boost compressor, and then regeneratively heating it before it enters the gasifier.

For the oxygen-blown cases an air-separation plant produces 98-percent-pure oxygen at atmospheric pressure and temperature. The oxygen was then compressed with a motor-driven, intercooled compressor and reheated using intercooler heat to match the oxygen inlet conditions for which the gasifier data apply.

The mass ratios used for the five gasifiers and the fuel gas composition before and after the cleanup process are shown in appendix A. It was assumed that the cleanup system removed 95 percent of the H_2S and 100 percent of any NH_3 present in the raw gas. Currently, it is uncertain which of the NSPS would apply to the pollutants emitted from IGCC plants. The possibilities include the NSPS for coal-fired steam-turbine electric generating units and the NSPS for stationary gas-turbine generating units. The more stringent standard for SO_x is in the NSPS for coal-fired electric generating units, which require a 90-percent sulfur removal for an Illinois No. 6 type coal. The SO_x emissions considered here are lower than either of these two standards and could be reduced further using presently available technology for relatively little increase in cost.

A compressor pressure ratio of 10 was chosen for the cases using a 1365 K (2000° F) gas-turbine firing temperature. For 1700 K (2600° F) and 1920 K (3000° F) gas-firing temperatures, a compressor pressure ratio of 16 was chosen. Air convection cooling was assumed at 1365 and 1700 K (2000° and 2600° F) firing temperatures. Three advanced cooling techniques were used at the 1920 K (3000° F) firing temperature. The turbine-cooling methods used for each firing temperature are described in appendix B.

For gas-turbine firing temperatures of 1700 and 1920 K (2600° and 3000° F), the steam-turbine throttle pressure and temperature were 16.6 MPa (2400 psia) and 810 K (1000° F), respectively, with a single reheat to 810 K (1000° F). The steam throttle conditions used at the 1365 K (2000° F) firing temperature were 10.0 MPa (1450 psia) and 785 K (950° F), with a single reheat to 785 K (950° F).

A detailed description of the cases presented in figures 1 to 5 is given in appendix A.

PERFORMANCE AND COMPARISONS

The results of this analysis will be discussed in two parts. First, a performance comparison will be made between IGCC systems using the five gasification concepts at a gas-turbine firing temperature of 1700 K (2600° F). Following this will be a comparison showing the effect of gas-turbine firing temperature on the IGCC system performance for three gasifier concepts. The range of gas-turbine firing temperatures selected reflects a range from present state-of-the-art gas-turbine technology to advanced, high-temperature technology, which requires use of the advanced turbine-cooling techniques now in development.

Comparison of IGCC Systems Using Five Gasifier Concepts

The performance of IGCC systems using the five gasifier concepts for a gas-turbine firing temperature of 1700 K (2600° F) is summarized in table IV. The variation between the highest and lowest gross electrical output of the five cases is about 6 percent. However, the power split between the gas and steam turbines varies considerably. The gas turbine of case WA17CB produces 53.5 percent of the gross electrical output, compared with 45.2 percent for case TA17CB for IGCC systems using air-blown gasifiers. In the oxygen-blown systems, the gas turbine for case BO17CB produces 65 percent of the gross electrical output, compared with 58.7 and 54.4 percent for cases WO17CB and TO17CB, respectively. The power splits for the systems considered here depend on the relative amounts of chemical and thermal energy in the fuel gas. The larger the fraction of fuel gas energy, which can be utilized directly in the gas turbine, the greater the gas-turbine contribution. The relative amounts of chemical and thermal energy for each system will be discussed later in this section.

The auxiliary power requirements for all five cases are approximately the same, except for the power required by the oxidant supply system. The oxidant supply power requirements are considerably greater for both the air- and oxygen-blown IGCC systems using the Texaco gasifiers (cases TA17CB and TO17CB) compared with their respective counterparts. The higher oxidant auxiliary power requirement for case TA17CB, compared with case WA17CB, is due to a 47-percent greater oxidant flow rate per unit of coal (appendix A) into the gasifier. The auxiliary power requirement for case TO17CB is greater than either WO17CB or BO17CB because of both increased oxidant flow per unit of coal and a higher gasifier operating pressure.

Case BO17CB has the highest net system efficiency of the five IGCC cases. The higher efficiency is partly due to a higher fraction of the fuel gas energy being used in the gas turbine and partly because the gasifier requires less oxidant per unit of coal, which reduces the auxiliary power requirements. The net system efficiency ranges from a high of 42.1 percent for case BO17CB to 39.3 percent for case WO17CB. There is little difference between the Texaco air- and oxygen-blown cases (TA17CB and TO17CB) or between the Westinghouse air- and oxygen-blown cases (WA17CB and WO17CB). In these cases the advantage of having a higher fraction of fuel gas energy used by the gas turbine for the oxygen-blown cases is offset by a higher auxiliary power to supply the oxygen. For these particular operating parameters, system efficiency will not be the sole discriminator of those IGCC systems; other factors will be important, including cost, NO_X control, load following capability, etc.

The interactions between the various IGCC subsystem are depicted by the generalized energy flow diagram shown in figure 6. This energy flow diagram, representative of each of the IGCC systems analyzed in this report, consists of essentially six basic subsystems for the air-blown cases and seven for oxygen-blown cases. The six basic air-blown subsystems are (1) the gasifier, which includes coal handling and preparation; (2) the raw gas cooler; (3) the cleanup system; (4) the gas turbine; (5) the heat-recovery steam generator; and (6) the steam turbine. The oxygen-blown cases, in addition to the six subsystems listed above, also include an air-separation plant.

In the simplified diagram of figure 6, only the major energy flow streams are indicated. Raw-fuel-gas energy Q_{rg} leaves the gasifier in the form of chemical Q_{ch} and thermal Q_{th} energy. A portion of the energy in the raw gas is input to the gas-turbine subsystem Q_{gt}. A second portion is used to generate steam for the steam turbine. A third portion is used to raise steam for the gasifier (not shown). And a fourth portion is lost in the raw gas cooler and cleanup systems. A large portion of the energy in the gas turbine exhaust fQ_{exh} is used to raise steam for the steam turbine. The remaining energy used to raise steam comes from cooling the air or water used in turbine cooling Q_m, and in some cases, from the gasifier Q_g. In the case of the BGC, gasifier Q_g represents steam raised in cooling the gasifier jacket, and in the Westinghouse gasifier cases, Q_g represents steam raised in cooling the raw gas that is recycled back to the gasifier.

Some of the more significant energy ratios, based on the generalized energy diagram shown in figure 6, are compared in table V for the five IGCC systems using the various gasifier concepts. These parameters were chosen because they appear in the relationship for system efficiency developed in appendix C.

The energy ratios listed in table V are grouped into four general categories dealing with the gasifier subsystem, the interface between the gasifier subsystem and the combined cycle, the combined cycle itself, and finally the entire IGCC system.

Gasifier subsystem. - The term "coal-gas efficiency" (the ratio of raw-gas chemical energy to coal chemical energy) is sometimes used to evaluate the performance of gasifiers. The term, however, is not appropriate when used as a figure of merit in determining the gasifier performance when the gasifier is integrated with combined cycles. Case W017CB, which has the next to the highest gasifier coal-gas efficiency, actually has the lowest net system efficiency. The coal-gas efficiency neglects several important gasifier energy inputs, such as those of the steam and oxidant requirements. It also does not include such outputs as the thermal energy in the raw gas or any steam generated within the gasifier.

The relative amounts of chemical and thermal energy in the raw gas leaving the gasifier for the given IGCC systems vary widely. The chemical energy portion of the raw gas energy Q_{ch}/Q_{rg} ranges from the high of 96 percent for case B017CB to 62 percent for case TA17CB.

Gasifier/combined cycle interface. - As mentioned previously, the larger the fraction of fuel energy used directly by the gas turbine, the larger its contributin to the total system power. Approximately 95 percent of the energy in the raw gas is used directly in the gas turbine Q_{gt}/Q_{rg} in case B017CB. This energy split resulted in the gas turbine producing 65 percent of the gross system power. Case TA17CB, which had the lowest percentage (66 percent) of fuel gas energy used directly in the gas turbine, had 45 percent of the gross power produced by the gas turbine.

The total fuel energy to the gas turbine consists of chemical and thermal energy. The chemical energy portion is dependent on the particular fuel-gas composition produced by the gasification/cleanup systems. The percent of the total chemical energy in the fuel gas used by the gas turbine a exceeds 92

percent for all cases. The thermal energy portion is dependent on the fuel-gas temperature entering the combustor, which for the cases presented was assumed to be 590 K (600° F), as well as on the fuel-gas composition.

The fraction of the thermal energy in the fuel gas used by the gas turbine y ranges from 0.136 for case T017CB to 0.565 for case B017CB. Although not considered in this analysis, the thermal portion could be increased by increasing the temperature of the fuel going to the gas-turbine combustor. However, where fuel and air are premixed before combustion (for NO_x control), autoignition and flashback considerations would limit fuel temperature to approximately 810 K (1000° F).

The fuel-gas thermal energy not used by the gas turbine can be used to raise additional steam to increase the power produced by the steam turbine. Approximately 79 percent of the thermal energy in raw fuel gas x was used to raise additional steam for the steam turbine used with the Texaco gasifiers. For the IGCC systems using the Westinghouse gasifiers, between 43 and 59 percent of the thermal energy was used to raise steam. (These values do not include steam raised by the fuel gas that is cooled and recycled back to the gasifier.)

Combined cycle. - Direct comparison of the gas-turbine subsystem efficiency η_{gt} between the air- and oxygen-blown cases is difficult because of the method used to account for the power required to deliver the oxidant to the gasifier. For the air-blown cases the oxidant is supplied by the gas-turbine compressor. The gas-turbine subsystem efficiencies shown for the air-blown cases include this additional compressor power which amounts to between 6 and 10 percent of the total energy in the fuel. The gas-turbine subsystem efficiency for case TA17CB is less than case WA17CB, primarily because of larger gasifier air requirements per unit of coal input. The gas-turbine subsystem efficiency for all three oxygen-blown cases is almost the same. The gas-turbine subsystem efficiency values shown for the oxygen-blown cases do not include the power required to supply the gasifier oxidants (this power is included in auxiliaries) and, therefore, are higher than those for the air-blown cases.

The steam-turbine subsystem efficiencies obtained for the five cases are consistent with the high steam-throttle and reheat conditions used. Proper selection of steam-turbine operating conditions is an important factor in IGCC system performance, even in cases which have low fractions of thermal energy in the fuel gas. With the exception of case TA17CB, the majority of the energy input to the steam cycle (over 67 percent) was supplied by the HRSG. For case TA17CB the energy supplied to the steam cycle is almost equally split between the raw gas cooler (RGC) and HRSG. The steam turbine subsystem efficiency for cases W017CB and B017CB is slightly lower than that for the other cases because steam induction was used. In these two cases additional thermal energy was available to raise low-pressure steam for induction into the low-pressure stages of the steam turbine. Even though this steam induction slightly decreases steam-turbine subsystem efficiency, it increases the amount of thermal energy recoverable, which results in increased steam-turbine power and system efficiency.

As mentioned previously, a raw gas cooler was not used in case B017CB because of the low temperature of the gas exiting the gasifier. A small amount

of saturated steam (0.136 kg/hr of steam per kg/hr of coal) is raised in the water jacket surrounding the gasifier and added to the exhaust flow of the high-pressure turbine.

The net system efficiencies are shown in table V. The relatively large fraction of auxiliary power required for case T017CB reduces the system efficiency significantly compared with the other cases.

A more detailed description of the interaction between the various subsystems of the five IGCC systems is depicted by the energy flow diagrams shown in appendix D. These diagrams show the manner in which the chemical energy in the coal is distributed and converted to other forms of energy as it flows through each of the major subsystems. The energy values are given in megawatts. They indicate the sum of the chemical and thermal energy of flow streams (MW_t) or electric power produced or required by subsystems. Table X of appendix D presents the energy ratios associated with each of the major subsystems (figs. 9 to 13 of appendix D).

Comparison Based on Gas Turbine Firing Temperatures

Nine additional cases were configured to determine the effect of gas-turbine firing temperature on IGCC system performance. Improving gas-turbine performance is one method of improving IGCC system performance. Gas-turbine efficiency can be improved by increasing the firing temperature. Increased firing temperatures require advanced cooling techniques which introduce performance penalties. The overall system efficiency would improve if these penalties did not offset the potential gain caused by increased firing temperature.

The results for these additional cases are presented in tables VI(a) to (c). This table presents comparisons for firing temperatures of 1365, 1700, and 1920 K (2000, 2600, and 3000° F) for the Westinghouse air- and oxygen-blown gasifiers and the BGC slagging gasifier, respectively. The respective results for the 1700 K (2600° F) firing temperature (previously presented) are shown for comparison. In Tables VI(a) and (b) results are also presented for the advanced convection and transpiration turbine-cooling methods for the 1920 K (3000° F) firing temperatures. In table VI(c) results are presented for advanced convection and water turbine cooling for the 1920 K (3000° F) firing temperature.

The net system efficiencies shown in table VI are plotted in figure 7. In all cases, performance increased with increased firing temperatures. Increasing the firing temperature from 1365 to 1700 K (2000 to 2600° F) (for convection cooled turbines) increased the efficiency by 7.1, 8.6, and 8.8 percent for the cases using the Westinghouse air- and oxygen-blown and the BGC gasifiers, respectively. The percent increase in efficiency is significantly less going from a 1700 to a 1920 K (2600 to 3000° F) firing temperature, especially for the air-blown gasifier. Here, the percent increase is 2.5, 4.8, and 6.2 for the Westinghouse air- and oxygen-blown and the BGC gasifiers, respectively.

The BGC gasifiers had the highest performance over the range of firing temperatures considered. The advantage of this gasifier configuration over the next highest performance (Westinghouse air-blown gasifiers) increased for

higher firing temperatures. The Westinghouse air-blown gasifier has a 1.9-point net efficiency advantage over its oxygen-blown counterpart at a 1365 K (2000° F) firing temperature, but this advantage decreases for increased firing temperatures until at 1920 K (3000° F) it is only 0.6 point higher for either of the two advanced turbine-cooling methods. As discussed previously, the system efficiencies for the Texas air- and oxygen-blown gasifier configurations are nearly the same at a firing temperature of 1700 K (2600° F) and fall between the Westinghouse airand oxygen-blown gasifier configurations.

Also illustrated in figure 7 is a comparison of advanced high-temperature cooling methods. The use of level-C cooling resulted in a 1.4-percentage point higher system efficiency than use of water cooling for the IGCC system fueled by the BGC slagging gasifier. Water cooling resulted in a slightly higher gas turbine power output than that obtained with level C because of the absence of compressor bleed air for cooling. This advantage, however, is offset by the large heat loss from the gas path to the low-temperature airfoils which lowers the gas turbine exhaust temperature, reducing the amount of thermal energy recoverable by the steam cycle. The reduced thermal energy input to the steam cycle lowers the steam-turbine power, which reduces the net system efficiency.

Level-C cooling also has a slight advantage (0.2 percentage point) over transpiration cooling in both the air- and oxygen-blown IGCC systems using the Westinghouse gasifier. The high level of cooling effectiveness with transpiration cooling results in higher gas turbine exhaust temperatures, which increases the thermal energy transferred to the steam cycle; thereby increasing the steam turbine power output. However, the increase in steam turbine power is balanced by a decrease in gas turbine power output due to the aerodynamic losses associated with transpiration cooling.

Major Subsystem Parameters and Their Effect on Performance

A set of influence coefficients was developed to provide a general understanding of the effect that certain system parameters have on system efficiency. Comparisons of these coefficients could identify the subsystems or operating parameters which, through improved technology, would result in greater improvement in total system performance. The development of these influence coefficients is described in appendix C. The influence coefficients (eqs. (C2) to (C7)), were obtained by partial differentiation of a simplified system efficiency relation with respect to each of six system parameters. An inherent assumption in this development is that these system parameters are independent; that is, one parameter can be changed without affecting the values of the others. For some of the parameters selected this is a reasonable assumption. For others there is a coupling between parameters where a change in one requires a change to one or more of the others. The influence coefficients for these parameters would only indicate a partial sensitivity.

The influence coefficients are presented in table VII for the IGCC system using the five gasifiers at a firing temperature of 1700 K (2600° F). For three out of the five IGCC systems (WA17CB, TA17CB, and TO17CB), the system efficiency is most sensitive to changes in steam-turbine subsystem efficiency, followed closely by gas-turbine subsystem efficiency. In these cases a 10-percent increase in steam-turbine efficiency would result in system efficiency increases between 5 and 6 percent. For case BO17CB, where 95 percent of the

fuel energy is used directly in the gas turbine, a 10-percent increase in gas-turbine subsystem efficiency would result in a 5-percent increase in system efficiency. These influence coefficients indicate that it is equally important to use the highest technology possible for both the steam and gas-turbine subsystems.

For the cases presented in this analysis, high steam-throttle conditions were used (consistent with heat-exchanger temperature limitations) along with steam reheat and, where possible, steam induction to obtain the maximum steam turbine power from the recoverable thermal energy. The system efficiency would be significantly reduced, for example, if the steam conditions were lowered because of economic considerations or if heat-exchanger design limitations prevented superheating in the raw gas cooler.

Another relatively sensitive parameter is the fraction f of the thermal energy available in the HRSG that actually is recovered by the steam cycle. As can be seen from the influence coefficient and from equation (C1), it is important not only to configure the steam cycle for high steam-cycle efficiency, but to also maximize the product of f and η_{st}, in order to maximize the total system efficiency.

The system efficiency sensitivity to changes in the fraction of raw-gas thermal energy that is used directly in the gas turbine y is low because the analysis does not consider the effect of coupling with other system parameters. Table VII indicates that a 100-percent increase in y would result in less than a 1-percent increase in system efficiency. An increase in y entails an increase in fuel-gas temperature to the gas-turbine combustor. An increase in the fuel temperature has a positive effect on gas-turbine performance. An increase in y reduces the amount of raw-gas thermal energy that is available to raise steam for the steam turbine, which may also affect steam-turbine performance.

Neither of these coupling effects were included in the analysis. Fuel temperature to the combustor could also be increased through the use of a medium or hot gas cleanup system. Reference 12 indicates a 3.6-percent increase in IGCC system efficiency going from a cold to a hot gas cleanup system, and from a fuel temperature of 705 to 865 K (810° to 1100° F) into the combustor (approximately a 40-percent increase in y). However, this increase also included the effects of changes to other system parameters as well as a change to the system configuration. As mentioned previously, there is an upper limit of (~810 K (1000° F)) to the fuel-gas temperature into the combustor (in combustors which premix the oxidant and fuel for NO_x control) before autoignition and flashback become problems.

Table VII also indicates the importance of minimizing the auxiliary power requirements and also of using all of the available thermal enregy.

CONCLUDING REMARKS

An investigation was conducted to compare the performance of IGCC systems using alternative types of gasifiers over a range of gas-turbine firing temperatures. Three types of coal gasification systems were considered: (1) entrained-bed gasifiers (both air- and oxygen-blown), represented by the Texaco

gasifier concept, (2) fluid-bed gasifiers (both air- and oxygen-blown), represented by the Westinghouse concept, and (3) an oxygen-blown, advanced, fixed-bed gasifier, represented by the British Gas Corporation (BGC) concept.

Five cases were configured using these gasifier, oxidant combinations for a gas-turbine firing temperature of 1700 K (2600° F). The gas turbine was assumed to use an advanced convection-cooling method. Each of the five cases used steam-turbine throttle conditions of 16.55 MPa and 810 K (2400 psia and 1000° F) with a single reheat to 810 K (1000° F). The gasifier input requirements and raw-fuel-gas composition and flows were taken from information presented in the literature. Some of these IGCC cases were then modified to evaluate the effects of a 1365 and a 1920 K (2000° and 3000° F) gas-turbine firing temperature.

The results indicated that for the conditions studied, the IGCC system using the BGC slagging gasifier had the highest net system efficiency (42.1 percent). The other four gasifier configurations had system efficiencies of approximately 40.0 percent. Based on these results, it appears that system efficiency will not be the sole discriminator in the selection of an IGCC configuration.

The distribution of chemical (higher heating value) versus thermal (sensible + latent heat) energy in the fuel gas determines the gas-turbine/- steam-turbine power split. All of the available thermal energy, in both the raw gas cooler and HRSG, with the exception of that used to generate gasifier steam and reheat the fuel gas, was used in the production of steam for the steam turbine cycle. Superheating was done in the raw gas cooler, but as much of the steam cycle boiling duty as possible was done in the raw gas cooler to minimize the hot end-metal temperatures. Superheating in the raw gas cooler could be a difficult design task and high risk area, especially for the extremely high gasifier exit temperatures from the Texaco gasifiers.

The BGC gas slagger fuel gas contained the highest fraction of chemical energy (~0.96), and the air-blown Texcaco gasifier the lowest fraction (0.62) of the five configurations. The fuel gas from the Texaco gasifier had the lowest fraction of chemical energy (~0.80) of the oxygen-blown configurations. The ratio of gas-turbine power to steam-turbine power follows the same trend from a high of 1.88 for the BGC slagger to 0.83 for the air-blown Texaco gasifier.

A comparison of the results of selected IGCC systems at gas-turbine firing temperatures of 1365 and 1920 K (2000° and 3000° F) with those at 1700 K (2600° F) shows increased system efficiency for increased firing temperature. The net efficiency for increased firing temperature was greatest for the IGCC systems using the BGC slagging gasifier with its higher chemical energy content fuel gas. The air-blown version of the fluid-bed gasifier (Westinghouse) has an efficiency advantage over its oxygen-blown counterpart for the three firing temperatures considered. However, the advantage decreases as firing temperature increases, such that at 1920 K (3000° F) there is only a 0.6-point advantage.

IGCC system performance was compared at the 1922 K (3000° F) firing temperature for three cooling technologies: advanced convection, transpiration, and water cooling. The use of the advanced convection-cooling technology leads to

higher IGCC system performance than either water cooling with a BGC gas slagger gasifier or transpiration cooling with a Westinghouse gasifier. The use of water cooling results in higher gas-turbine power output because of the absence of compressor bleed air for cooling. But this advantage is offset by the large heat losses from the gas path to the low-temperature airfoils. These heat losses lower the turbine-exhaust-gas temperature and reduce the heat recoverable by the steam cycle, thus reducing steam-turbine power. The opposite is true for transpiration cooling. Transpiration cooling has a higher effectiveness than the advanced convection cooling, which results in higher gas-turbine-exhaust temperatures, which ultimately result in increased steam-turbine power output. The increased steam power, however, is offset by a decrease in gas-turbine power output due to the aerodynamic losses associated with transpiration cooling.

A simplified analysis was performed to identify the particular parameters most influential in determining net system performance. A set of sensitivity parameters was developed using the results from the five cases which compared IGCC systems with different gasifiers. For the five cases for which these sensitivity parameters were evaluated, system efficiency is most sensitive to changes in steam- and gas-turbine subsystem efficiencies. System efficiency is more sensitive to changes in steam-turbine efficiency in the cases where sizable fractions of the raw gas thermal energy were used to raise steam for the steam turbine.

The analysis also indicated that system efficiency was relatively insensitive to increases in the amount of raw-gas thermal energy that is used directly in the gas turbine (through increased fuel temperature to the gas turbine combustor). However, the analysis is only approximate for this parameter in that it does not consider any coupling between it and the gas- and steam-turbine efficiencies.

For the cases considered in this report, IGCC system efficiency can be increased equally well by improving the gas- or steam-turbine subsystem performance. Gas-turbine performance can be improved through the use of higher firing temperatures, provided that the required turbine-cooling technology does not overpenalize turbine efficiency. Steam-turbine subsystem performance can be maximized through the judicious use of thermal energy to produce high steam throttle and reheat conditions.

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TABLE I. - SUMMARY OF FACTORS USED TO DETERMINE AUXILIARY POWER REQUIREMENTS

	Coal preparation and feed, kW_e / kg coal/hr	Oxygen plant[a], kW_e / kg oxygen/hr	Boost air compressor, MW_e / kg air/hr	Gas production[b], kW_e / kg coal/hr	Cleanup system, kW_e / kg coal/hr	Boiler feed pump[c], MW_e	Steam turbine, MW_e / MW_e, st. turb.	Other electrical, MW_e / MW_e, tot out
Westinghouse air-blown	[d]0.023	-------	[e]2.784×10^{-5}	[d]0.015	[f]0.036	[g]6.40	[d]0.020	[d]0.015
Texaco air-blown	[h]0.029	-------	[e]2.784×10^{-5}	[h]0.013	[f]0.036	7.79	[d]0.020	[d]0.015
Westinghouse oxygen-blown	[d]0.023	[i]21.435	-----------	[d]0.015	[f]0.036	5.53	[d]0.020	[d]0.015
Texaco oxygen-blown	[h]0.029	[i]22.597	-----------	[h]0.013	[f]0.036	6.88	[d]0.020	[d]0.015
BGC oxygen-blown	[h]0.016	[i]21.435	-----------	[h]0.014	[f]0.036	4.36	[d]0.020	[d]0.015

[a]Includes air separation plant and oxygen compression.
[b]Ash removal, gas cooling, recycle power, and process treatment.
[c]Calculated based on stream turbine parameters.
[d]NASA estimate.
[e]Reference 1.
[f]Reference 7.
[g]Reference 6.
[h]Reference 8.
[i]Reference 9.

TABLE II. - OPERATING PARAMETERS

	Case designation						
	WA13CA	WA17CB	WA19CC	WA19T	W013CA	W017CB	W019CC
Coal input, kg/hg (lb/hr)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)
Gasifier outlet temperature, K (°F)	1285 (1850)	1285 (1850)	1285 (1850)	1285 (1850)	1285 (1850)	1285 (1850)	1285 (1850)
Gasifier operating pressure, MPa (psia)	2.9 (415)	2.9 (415)	2.9 (415)	2.9 (415)	2.9 (415)	2.9 (415)	2.9 (415)
Fuel gas to combustor, K (°F)	590 (600)	590 (600)	590 (600)	590 (600)	590 (600)	590 (600)	590 (600)
Gas turbine firing temperature, K (°F)	1365 (2000)	1700 (2600)	1920 (3000)	1920 (3000)	1365 (2000)	1700 (2600)	1920 (3000)
Gas turbine exhaust temperatuare, K (°F)	830 (1037)	895 (1155)	1035 (1402)	1065 (1458)	830 (1037)	895 (1155)	1035 (1403)
Compressor pressure ratio	10	16	16	16	10	16	16
Stack gas temperature, K (°F)	420 (300)	420 (300)	420 (300)	420 (300)	420 (300)	420 (300)	420 (300)
Steam turbine throttle conditions:							
Pressure, MPa (psia)	10.0 (1450)	16.6 (2400)	16.6 (2400)	16.6 (2400)	10.0 (1450)	16.6 (2400)	16.6 (2400)
Temperature, K (°F)	785 (950)	810 (1000)	810 (1000)	810 (1000)	785 (950)	810 (1000)	810 (1000)
Reheat temperature, K (°F)	785 (950)	810 (1000)	810 (1000)	810 (1000)	785 (950)	810 (1000)	810 (1000)
	W019T	TA17CA	T017CB	B013CA	B017CB	B019CC	B019W
Coal input, kg/hr (lb/hr)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)	160 500 (353 900)
Gasifier outlet temperature, K (°F)	1285 (1850)	1590 (2400)	1590 (2400)	710 (820)	710 (820)	710 (820)	710 (810)
Gasifier operating pressure, MPa (psia)	2.9 (415)	2.9 (415)	5.1 (735)	2.9 (415)	2.9 (415)	2.9 (415)	2.9 (415)
Fuel gas to combustor, K (°F)	590 (600)	590 (600)	590 (600)	590 (600)	590 (600)	590 (600)	590 (600)
Gas turbine firing temperature, K (°F)	1920 (3000)	1700 (2600)	1700 (2600)	1365 (2000)	1700 (2600)	1920 (3000)	1920 (3000)
Gas turbine exhaust temperature, K (°F)	1065 (1459)	905 (1167)	900 (1156)	830 (1032)	890 (1146)	1030 (1391)	1005 (1347)
Compressor pressure ratio	16	16	16	10	16	16	16
Stack gas temperature, K (°F)	420 (300)	420 (300)	435 (319)	420 (300)	420 (300)	420 (300)	420 (300)
Steam turbine throttle conditions:							
Pressure, MPa (psia)	16.6 (2400)	16.6 (2400)	16.6 (2400)	10.0 (1450)	16.6 (2400)	16.6 (2400)	16.6 (2400)
Temperature, K (°F)	810 (1000)	810 (1000)	810 (1000)	785 (950)	810 (1000)	810 (1000)	810 (1000)
Reheat temperature, K (°F)	810 (1000)	810 (1000)	810 (1000)	785 (950)	810 (1000)	810 (1000)	810 (1000)

TABLE III. - SYSTEM DESCRIPTION CODE[a]

	Code	Description
Gasifier designation, XX	WA	Westinghouse air-blown gasifier
	WO	Westinghouse oxygen-blown gasifier
	TA	Texaco air-blown gasifier
	TO	Texaco oxygen-blown gasifier
	BO	British Gas Corp., oxygen-blown slagging gasifier
Gas turbine firing temperature, YY	13	1365 K
	17	1700 K
	19	1920 K
Turbine cooling technology, ZZ	CA	Convection air cooled, level A
	CB	Convection air cooled, level B
	CC	Convection air cooled, level C
	T	Transpiration cooling
	W	Water cooled

[a]System description given by XX YY ZZ. For example, WO17CB is the designation for the case for the Westinghouse oxygen-blown gasifier using a gas turbine firing temperature of 1700 K. The turbine is convectively cooled using the level B method.

TABLE IV. - IGCC SYSTEM PERFORMANCE SUMMARY

[Gas turbine firing temperature, 1700 K (2600° F).]

	Case designation				
	WA17CB	TA18CB	WO17CB	TO17CB	BO17CB
Coal input, MW_t	1268.7	1268.7	1268.7	1268.7	1268.7
Gas turbine generator output, MW_e	300.6	251.5	329.9	321.8	384.2
Steam turbine generator output, MW_e	261.6	304.6	232.3	270.0	204.0
Gross electrical output, MW_e	562.2	556.1	562.2	591.8	588.2
Auxiliary power requirements:					
Oxidant fupply system, MW_e	13.6	20.0	34.2	51.5	26.9
Coal preparation and feed, MW_e	3.5	4.5	3.5	4.5	2.4
Gas production, MW_e	2.4	2.0	2.4	2.0	2.0
Cleanup system, MW_e	5.5	5.5	5.5	5.5	5.5
Boiler feed pump, MW_e	6.4	7.5	5.5	6.6	4.4
Steam turbine auxiliaries, MW_e	5.2	6.1	4.7	5.4	4.1
Other electrical, MW_e	8.4	8.3	8.4	8.9	8.8
Total auxiliaries	45.0	53.9	64.2	84.4	54.1
Net electrical output, MW_e	517.2	502.2	498.0	507.4	534.1
Net system efficiency (net electrical/coal input), percent	40.8	39.6	39.3	40.0	42.1

TABLE V. - ENERGY RATIO COMPARISON

Energy ratio parameter	Case designation				
	WA17CB	TA17CB	WO17CB	TO17CB	BO17CB
Gasifier subsystem:					
Raw fuel gas chemical energy/coal input, Q_{ch}/Q_c	0.785	0.667	0.863	0.789	0.959
Raw fuel gas thermal energy/coal input, Q_{th}/Q_c	0.208	0.402	0.139	0.203	0.046
Raw fuel gas total energy/coal input, Q_{rg}/Q_c	0.993	1.069	1.002	0.992	1.004
Gasifier - combined cycle interface:					
Raw fuel energy to gas turbine/coal input, Q_{gt}/Q_c	0.807	0.710	0.826	0.797	0.956
Raw fuel energy to gas turbine/raw fuel gas total energy, Q_{gt}/Q_{rg}	0.813	0.664	0.824	0.803	0.952
Raw fuel thermal energy to steam turbine/coal input, xQ_{th}/Q_c	0.122	0.316	0.060	0.162	0
Raw fuel thermal energy to steam turbine/raw fuel gas total energy, xQ_{th}/Q_{rg}	0.123	0.296	0.060	0.163	0
Fuel gas chemical energy to gas turbine/raw fuel gas chemical energy, a	0.963	0.970	0.926	0.975	0.975
Fuel gas thermal energy to steam turbine/raw fuel gas thermal energy, x	0.589	0.787	0.430	0.796	0
Fuel gas thermal energy to gas turbine/raw fuel gas thermal energy, y	0.248	0.157	0.189	0.136	0.565
Combined cycle:					
Raw fuel gas thermal energy to steam/turbine/total energy to steam turbine, xQ_{th}/Q_{st}	0.246	0.485	0.130	0.268	-----
Gas gurbine exhaust energy to gas turbine/total energy to steam turbine, fQ_{exh}/Q_{st}	0.691	0.474	0.791	0.675	0.843
Miscellaneous energy to steam turbine/total energy to steam turbine, $(Q_g + Q_m)/Q_{st}$	0.063	0.041	0.079	0.057	0.157
Gas turbine efficiency, $\eta_{gt} \equiv P_{gt}/Q_{gt}$	0.293	0.279	0.315	0.318	0.317
Steam turbine efficiency, $\eta_{st} \equiv P_{st}/Q_{st}$	0.416	0.413	0.399	0.415	0.384
IGCC system:					
Auxiliary power/coal input, P_{aux}/Q_c	0.035	0.043	0.051	0.067	0.043
Net system efficiency, $\eta_{cc} \sigma P_{net}/Q_c$	0.408	0.396	0.393	0.400	0.421

TABLE VI. - COMPARISON OF PERFORMANCE RESULTS AT THREE GAS-TURBINE FIRING TEMPERATURES

(a) IGCC system using Westinghouse air-blown gasifier

	Case designation			
	WA13CA	WA17CB	WA19CC	WA19T
Coal input, MW_t	1268.7	1268.7	1268.7	1268.7
Gas turbine generator output, MW_e	283.3	300.6	307.9	301.4
Steam turbine generator output, MW_e	241.7	261.6	268.4	271.8
Gross electrical output, MW_e	524.0	562.2	576.3	573.2
Auxiliary power requirements, MW_e	41.2	45.0	45.5	45.6
Net electrical output, MW_e	482.8	517.2	530.8	527.6
Net system efficiency, percent	38.1	40.8	41.8	41.6

(b) IGCC system using Westinghouse oxygen-blown gasifier

	Case designation			
	WO13CA	WO17CB	WO19CC	WO19T
Coal input, MW_t	1268.7	1268.7	1268.7	1268.7
Gas turbine generator output, MW_e	304.5	329.9	337.6	330.7
Steam turbine generator output, MW_e	215.5	232.3	251.0	254.6
Gross electrical output, MW_e	520.0	562.2	588.6	585.3
Auxiliary power requirements, MW_e	60.7	64.2	65.5	65.6
Net electrical output, MW_e	459.3	498.0	523.1	519.7
Net system efficiency, percent	36.2	39.3	41.2	41.0

(c) IGCC system using British Gas Corporation slagging gasifier

	Case designation			
	BO13CA	BO17CB	BO19CC	BO19W
Coal input, MW_t	1268.7	1268.7	1268.7	1268.7
Gas turbine generator output, MW_e	354.1	384.2	393.9	395.1
Steam turbine generator output, MW_e	189.1	204.0	228.7	209.3
Gross electrical output, MW_e	543.2	588.2	622.6	604.4
Auxiliary power requirements, MW_e	51.6	54.1	56.1	55.0
Net electrical output, MW_e	491.5	534.1	566.5	549.4
Net system efficiency, percent	38.7	42.1	44.7	43.3

TABLE VII. - INFLUENCE COEFFICIENTS FOR IGCC SYSTEMS WITH A 1700 K GAS-TURBINE FIRING TEMPERATURE

Influence coefficient (a)	Case designation				
	WA17CB	TA17CB	WO17CB	TO17CB	BO17CB
$K_{\eta_{gt}}$	0.405	0.359	0.473	0.446	0.544
$K_{\eta_{st}}$	.506	.643	.466	.577	.399
K_f	.349	.288	.368	.359	.339
K_y	.007	.004	.006	.006	.005
$K_{(Q_m + Q_g)/Q_c}$	.032	.025	.037	.030	.060
$K_{(P_{aux}/Q_c)}$	-.087	-.109	-.130	-.168	-.101

[a]Defined in appendix C.

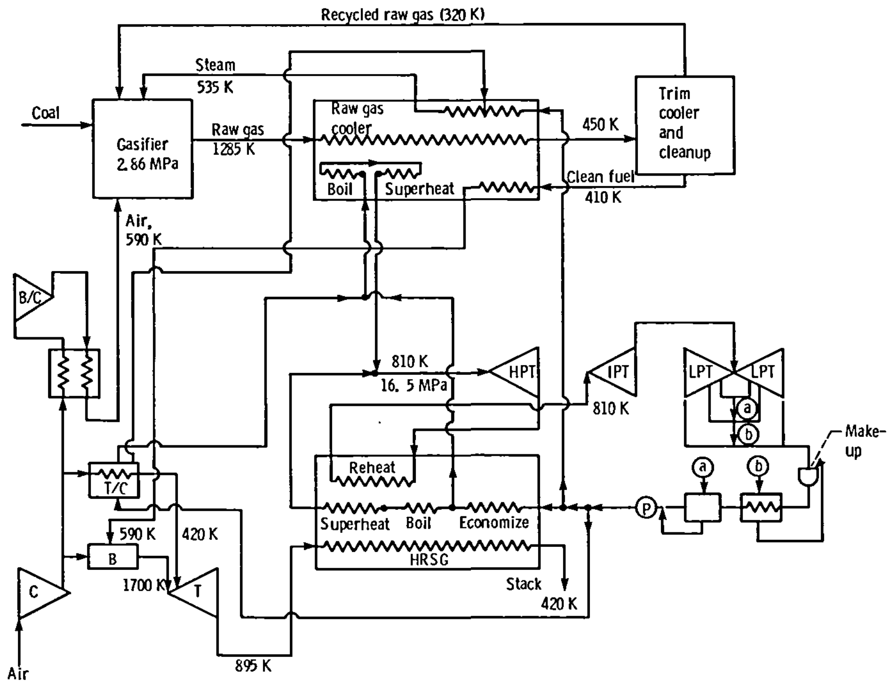

Figure 1. - Schematic of IGCC system using Westinghouse air-blown gasifier. B, burner; B/C, boost compressor; C, compressor; HPT, high pressure turbine; IPT, intermediate pressure turbine; LPT, low pressure turbine; P, pump; T, turbine; T/C, turbine cooler.

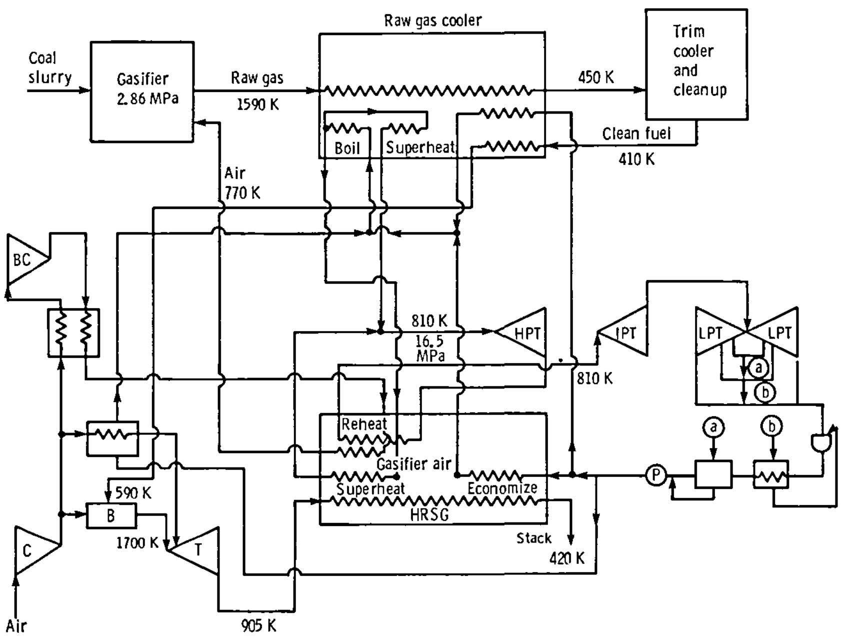

Figure 2. - Schematic of IGCC system using Texaco air-blown gasifier. See figure 1 for definitions of abbreviations.

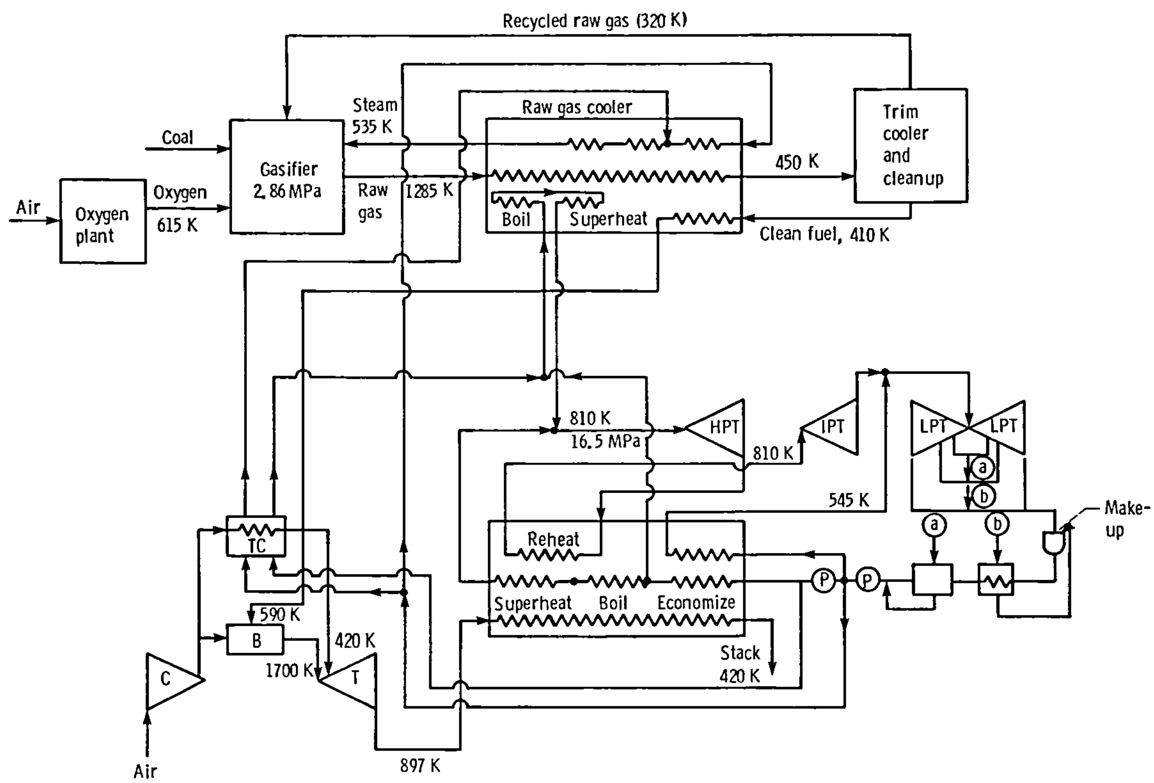

Figure 3. - Schematic of IGCC system using Westinghouse oxygen-blown gasifier. See figure 1 for definitions of abbreviations.

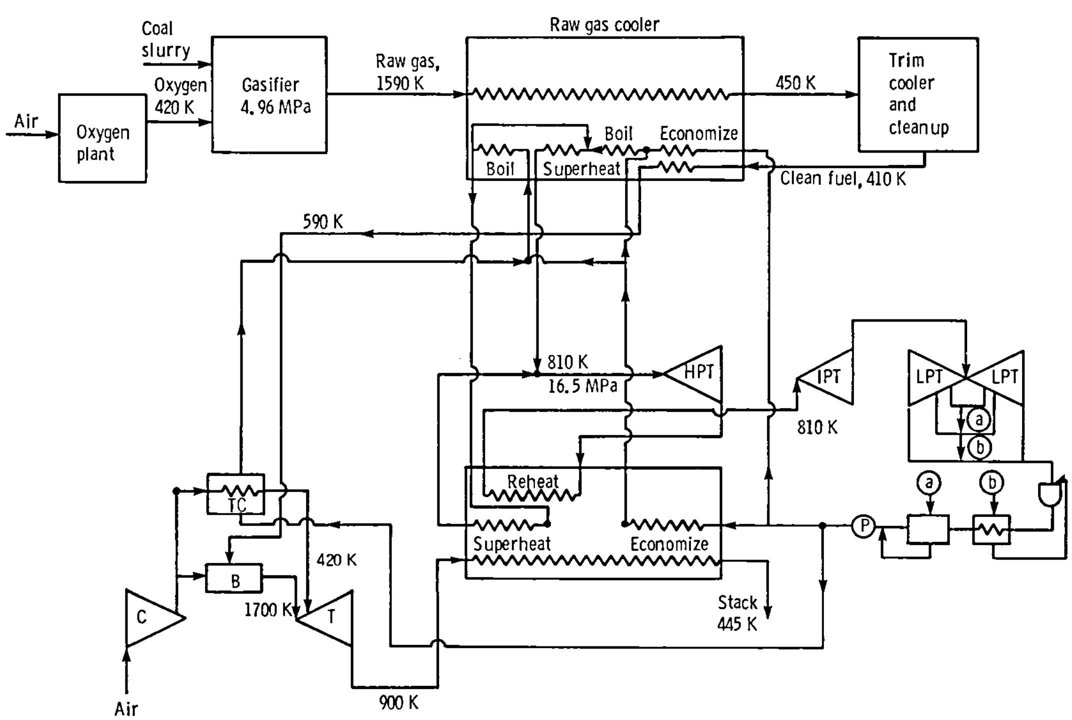

Figure 4. - Schematic of IGCC system using Texaco oxygen-blown gasifier. See figure 1 for definitions of abbreviations.

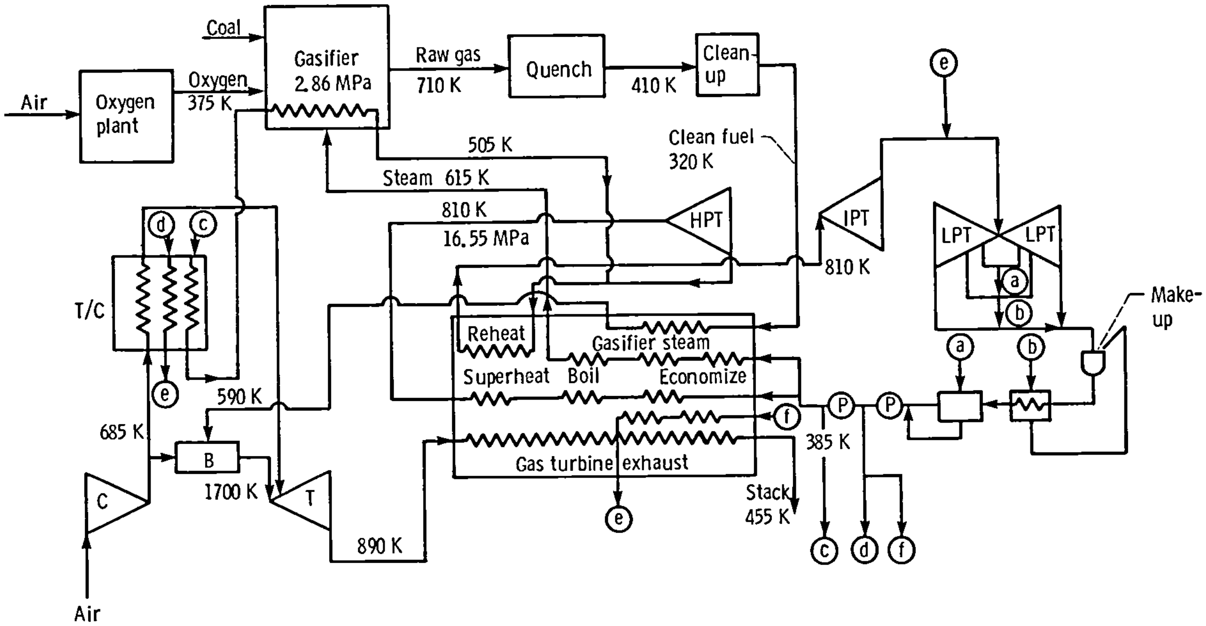

Figure 5. - Schematic of IGCC system using BGC slagger gasifier. See figure 1 for definitions of symbols.

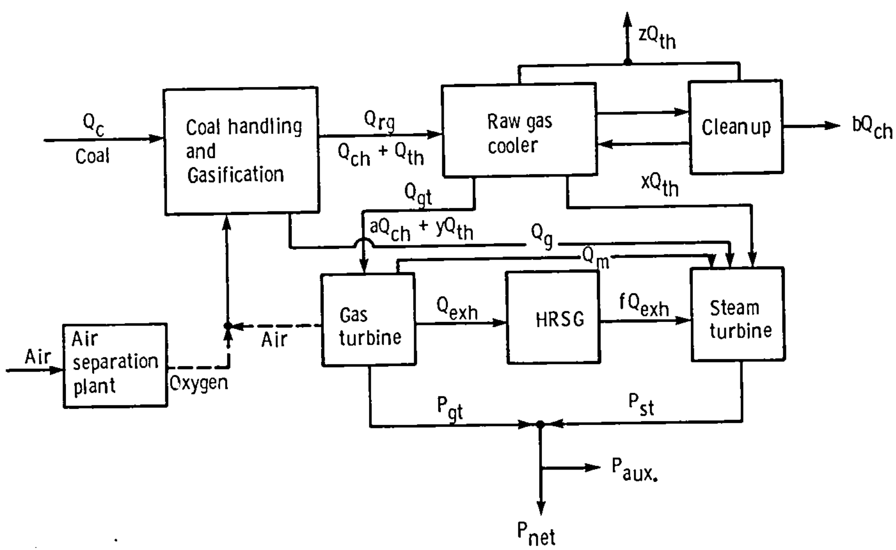

Figure 6. - Generalized energy flow diagram. (Symbols defined in appendix C.)

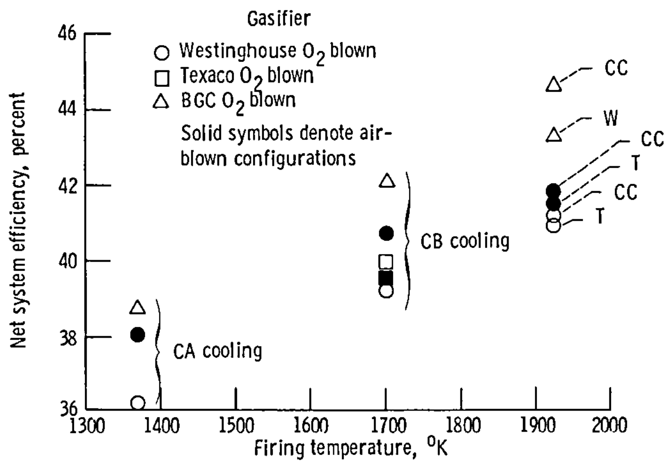

Figure 7. - Net IGCC system efficiency as a function of gas turbine firing temperature. (See table III for definitions of cooling terminology.)

APPENDIX A

DESCRIPTION OF IGCC SYSTEMS USING FIVE GASIFIER CONCEPTS AT A FIRING TEMPERATURE OF 1700 K (2600° F)

Case WA17CB - IGCC System Using the Air-Blown Westinghouse Gasifier

The basic gasifier operating parameters and raw-gas composition and flow rates were taken from data that were developed in a study reported in reference 7. The gasifier mass flow ratios are shown in table VIII. Coal and steam at 535 K (500° F) are partially combusted with 590 K (600° F) air to produce a 1285 K (1850° F) raw-fuel gas. The hot raw gas is successively cooled to 450 K (350° F) in the raw gas cooler (RGC) by generating and superheating part of the high-pressure steam used in the steam turbine, by generating and superheating the steam used in the gasifier, and by reheating the cleaned fuel to 590 K (600° F). The raw gas then enters the trim cooler where it is further cooled to approximately 315 K (110° F) by reheating clean fuel to 410 K (275° F). Approximately 7.0 percent of the raw gas is recycled back to the gasifier where it is used to help fluidize the char bed. The bulk of the raw gas enters the cleanup system where 95 percent of the hydrogen sulfide is removed. Information available in the literature suggests that the sulfur compounds in the coal are converted to hydrogen sulfide and carbonyl sulfide, which can be removed from the raw gas by several commercially proven acid-gas removal technologies. The cleaned fuel is then reheated in the trim cooler and raw gas cooler before entering the gas-turbine combustor. The fuel gas compositions at various stages along the process path are shown in table IX.

The advanced gas-turbine model used for all base cases is a 16:1 pressure-ratio machine with a combustor exit temperature of 1700 K (2600° F). The turbine is air cooled using advanced (level B) convection-cooling technology. The turbine-cooling technologies used in this analysis are discussed in appendix B. The compressor exit air used for turbine cooling is cooled to 420 K (300° F) by heating the feedwater for the steam turbine. The gas-turbine exhaust products are cooled from 895 to 420 K (1155° to 300° F) in the heat recovery steam generator (HRSG) by generating and superheating the remaining high-pressure steam-turbine throttle flow and by reheating all the steam flow to the intermediate-pressure steam turbine.

The steam-turbine throttle conditions were 16.55 MPa and 810 K (2400 psia and 1000° F) with a single reheat to 810 K (1000° F). The steam cycle used two stages of feedwater heating.

Case TA17CB - IGCC System Using the Air-Blown Texaco Gasifier

The basic gasifier operating parameters and raw-fuel gas composition were taken from reference 1. A slurry of coal and water (two parts dry coal to one part water) is partially combusted with 770 to 1590 (925° to 2400° F) raw-fuel gas. The high-level raw-gas thermal energy is used to evaporate all of the economized feedwater, which comes from the HRSG, from cooling the gas-turbine coolant, and from the raw gas cooler itself. High-level raw-gas thermal energy is also used to superheat a portion of the throttle steam. Low-level raw-gas

thermal energy is used to reheat the cleaned fuel to 590 K (600° F) and to economize a portion of the feedwater flow. Raw gas then enters the trim cooler at 450 K (350° F) where it is cooled to approximately 370 K (110° F) by regeneratively reheating the cleaned fuel to 410 K (275° F). The cooled raw gas then enters the cleanup system, where it is assumed that 95 percent of the hydrogen sulfide is removed. The fuel-gas composition at various stages along the process path is shown in table IX.

The basic gas-turbine operating parameters are the same as used for all configurations: a firing temperature of 1700 K (2600° F) and a compressor pressure ratio of 16:1. The main differences between configurations are the turbine exhaust mass flow and temperature; these differences are caused by the variations in fuel composition and chemical energy. The turbine is cooled using advanced (level B) convection cooling technology. The compressor exit air used for the turbine is cooled by economizing a portion of the steam cycle feedwater flow.

The gas-turbine exhaust products are cooled to 420 K (300° F) in the HRSG by superheating a portion and reheating all of the steam-turbine throttle flow in addition to economizing a portion of the feedwater flow and heating the gasifier air requirements to 770 K (925° F).

The steam-turbine throttle conditions were 16.55 MPa and 810 K (2400 psia and 1000° F) with a single reheat to 810 K (1000° F).

Case WO17CB - IGCC System Using the Westinghouse Oxygen-Blown Gasifier

The basic gasifier operating parameters and raw-fuel-gas composition were taken from data reported in reference 7. The operation of this configuration is basically the same as that for case WA17CB (fig. 1), except the oxidant is 98-percent-pure oxygen obtained from a separate air separation plant and compressed by a motor driven compressor to approximately 3.10 MPa (450 psia). The operation of the raw gas cooler, trim cooler, and cleanup system is the same as for WA17CB except that, in this case, approximately 20 percent of the cooled raw gas is required to be recycled back into the gasifier.

There are some variations between the oxygen- and air-blown versions in the way the HRSG and turbine cooler (T/C) were configured. Additional low-quality heat is available in the HRSG which was used to raise low-pressure steam for induction into the low-pressure turbine (LPT). Additional low-quality heat is also available in the cooling of the compressor discharge air that is used to cool the gas turbine. This additional heat is used to economize a portion of the low-pressure steam requirement of the gasifier.

Case TO17CB - IGCC System Using the Texaco Oxygen-Blown Gasifier

The basic gasifier operating parameters and raw-fuel-gas composition were taken from reference 1. Here, again, the basic configuration is nearly identical to its air-blown counterpart (fig. 2, p. 19), except for minor heat management techniques in the raw gas cooler. These variations in management techniques are necessary because of the much lower raw-gas mass flow per unit of coal input for the oxygen-blown gasifier. The relatively low raw-gas flow

places much more of the steam-raising duty in the HRSG. The chemical energy content in the clean fuel is over three times that of the air-blown case, which more than offsets the effect of its lower mass flow in terms of power-producing ability.

Case B017CB - IGCC System Using the British Gas Corporation Slagging Gasifier

The basic operating parameters and raw-fuel-gas composition were taken from reference 3. This configuration is considerably different from the other configurations. First, the gasifier has a water jacket which internally cools the gasified products to 710 K (820° F) before they exit the gasifier. The saturated steam raised in the water jacket is combined with the exhaust flow from the high-pressure stages of the steam turbine. The combined flow is then reheated to 810 K (1000° F) in the HRSG.

The raw gas out of the gasifier contains naphtha, tars, phenols, and other oils which are washed out in a water quench. The liquid hydrocarbons separated from the raw gas and water quench are mixed with coal fines from the coal preparation units and are recycled as a slurry to the gasifier and then to extinction. The raw-gas temperature is reduced to 410 K (280° F) by the water wash. This temperature was considered to be too low for practical use in raising steam, so a separate raw gas cooler was not used. The raw gas is sent to the cleanup system where 95 percent of the hydrogen sulfide is removed. The cleaned fuel is then reheated to 590 K (600° F) in the HRSG before entering the gas-turbine combustor. The fuel-gas composition at various stages along the process path is shown in table IX. The raw-gas composition shown exiting the gasifier does not include any of the liquid hydrocarbon products.

All of the throttle steam flow is raised in the HRSG, along with all of the reheat duty and the raising of the gasifier steam requirement.

The heat extracted from the compressor exit air used for turbine cooling was used to raise low-pressure steam for induction into the low-pressure stages of the steam turbine and to preheat the feedwater used in the gasifier cooling jacket.

TABLE VIII. - GASIFIER MASS RATIOS

Gasifier	Westinghouse	Texaco	Westinghouse	Texaco	BGC
Gasifier oxidant	Air	Air	Oxygen (98 percent)	Oxygen (98 percent)	Oxygen (98 percent)
Gasifier mass ratios:					
Recycles fines/raw gas	[a]0.8875	[b]0.8575	[a]0.6607	[b]0.8235	[c]0.9330
Coal/raw gas	.2237	.1718	.4041	.4560	.5991
Oxidant/coal	3.0420	4.4788	.5966	.8517	.4690
Water or steam/coal	.2499	.4790	.5001	.4790	.2983
Ash/coal	.1107	.0954	.1107	.0958	.0980
Recycled gas/raw gas	.0671	------	.2021	------	[d].0417
Fines out/raw gas	.0714	------	.1289	------	------
Recycles fines/raw gas	.9655	------	.9655	------	------

[a]Values taken from data that were developed in a study performed in reference 7.
[b]Values taken from reference 2.
[c]Values taken from reference 3.
[d]Recycled liquid hydrocarbons.

TABLE IX. - COMPARISON OF GAS COMPOSITION[a] BEFORE AND AFTER CLEANUP

[Location: 1, raw gas leaving gasifier; 2, raw gas entering cleanup system; 3, clean fuel.]

(a) Air-blown gasifiers

Fuel gas constituents	Location					
	1		2		3	
	Gasifier					
	Westinghouse	Texaco	Westinghouse	Texaco	Westinghouse	Texaco
	Mass fraction					
CO	0.2795	0.1908	0.2879	0.2085	0.2906	0.2098
CO_2	0.0878	0.1172	0.0905	0.1281	0.0913	0.1289
H_2	0.0111	0.0083	0.0115	0.0091	0.0116	0.0092
CH_4	0.0144	0.0006	0.0149	0.0007	0.0150	0.0007
N_2	0.5629	0.5791	0.5798	0.6327	0.5852	0.6368
H_2S	0.0093	0.0062	0.0096	0.0068	0.0005	0.0003
H_2O	0.0349	0.0864	0.0059	0.0018	0.0059	0.0018
COS	------	0.0010	------	0.0011	------	0.0011
A	------	0.0103	------	0.0113	------	0.0114
Higher heating valve, W/kg (Btu/lb)	5359 (2304)	3259 (1401)	5525 (2375)	3552 (1527)	5424 (2332)	3472 (1493)

(b) Oxygen-blown gasifiers

Fuel gas constituents	Location								
	1			2			3		
	Gasifier								
	Westinghouse	Texaco	BGC	Westinghouse	Texaco	BGC	Westinghouse	Texaxo	BGC
	Mass fraction								
CO	0.5252	0.5786	0.7811	0.5902	0.6896	0.8135	0.6034	0.7027	0.8375
CO_2	0.2498	0.1973	0.0409	0.2806	0.2352	0.0426	0.2869	0.2396	0.0439
H_2	0.0245	0.0282	0.0295	0.0275	0.0336	0.0307	0.0281	0.0343	0.0316
CH_4	0.0596	0.0016	0.0598	0.0670	0.0019	0.0623	0.0685	0.0020	0.0641
C_2H_4	------	------	0.0031	------	------	0.0032	------	------	0.0033
C_2H_6	------	------	0.0049	------	------	0.0051	------	------	0.0053
N_2	0.0044	0.0106	0.0074	0.0050	0.0126	0.0070	0.0051	0.0128	0.0079
H_2S	0.0205	0.0164	0.0226	0.0231	0.0196	0.0235	0.0011	0.0010	0.0012
H_2O	0.1160	0.1622	0.0430	0.0067	0.0015	0.0033	0.0068	0.0015	0.0034
COS	------	0.0026	0.0017	------	0.0031	0.0018	------	0.0032	0.0018
NH_3	------	------	0.0060	------	------	0.0062	------	------	------
A	------	0.0024	------	------	0.0029	------	------	0.0029	------
Higher heating value, W/kg (Btu/lb)	12 439 (5348)	10 243 (4404)	16 344 (7027)	13 970 (6007)	12 208 (5249)	17 016 (7316)	13 916 (5983)	12 127 (5214)	16 995 (7307)

[a]All values are mass fractions.

APPENDIX B

GAS-TURBINE COOLING TECHNIQUES

Higher firing temperatures are one means of increasing the efficiency of gas-turbine cycles. As the firing temperatures are increased, the turbine-cooling technology must improve to keep the metal temperatures below acceptable limits. The performance of the gas turbines reported herein was calculated for three combustor exit temperatures using several different cooling methods. These general cooling methods include convection-, transpiration-, and water-cooling techniques.

Three levels of convection cooling (designated levels A, B, and C) were used. Level A cooling technology is associated with current, state-of-the-art aviation gas turbine airfoils with firing temperatures to 1600 K (2420° F). These airfoils are either single-piece castings or two-piece castings bonded together (have about the same level of cooling performance as the single-piece casting). Level B cooling is applicable to firing temperatures to 1700 K (2600° F). Level B airfoils are multipiece castings having more complex internal cooling passages with smaller passage dimensions and wall thickness than current state-of-the-art designs. Level C cooling is applicable to firing temperatures up to 1950 K (3050° F). This cooling technology is based on the ultimate limit of multipiece airfoil fabrication techniques that could be achieved with further development.

Transpiration cooling is an advanced air-cooling technology which consists of an airfoil with a porous skin. The cooling air flows along internal channels and exits through the porous skin. The cooling air is thus an insulating layer between the airfoil and the hot gases. This cooling scheme is applicable to firing temperatures up to 1950 K (3050° F) and offers the potential for more effective cooling than the advanced convection techniques. Its disadvantage is that the introduction of coolant over the entire blade surface adversely affects the aerodynamic performance of the airfoils and allows the possibility of plugging of the porous skin, especially with the use of coal-derived fuels.

One of the cases considered used water to cool the hot turbine sections. The stators were cooled by a closed-loop, single-phase water system. The heat transferred to the water in cooling the stators was recovered for feedwater heating in the steam cycle. The rotor blades were cooled by an open-loop system, allowing boiling of the water coolant within the blades; the resultant water-steam mixture was discharged from the blade tip. Approximately one-third of this coolant is recovered while the rest is discharged to the gas path. The advantages of water cooling are that it eliminates the performance penalty associated with using compressor bleed air for cooling and it results in lower metal temperatures, thereby possibly lengthening the lives of those metal components. Its disadvantages are the large heat losses from the gas path to the lower temperature airfoils and the performance penalties associated with the discharge of the coolant into the gas path.

The amount of air cooling needed for each blade row was determined using the effectiveness curves shown in figure 8. The level A cooling curve approximates cooling data results for state-of-the-art convection cooling from reference 10. The curves for levels B and C convection cooling approximate those reported in reference 11. The curve for transpiration cooling was taken from

reference 10. An additional 1 percent of the compressor exit flow per turbine stage was assumed for wheelspace cooling. The maximum allowable metal temperature was assumed to be 1090 K (1500° F) for all air-cooled cases and 810 K (1000° F) for the water-cooled case. It was assumed that the maximum amount of air coolant for a single stator and for a blade row was 6 and 3-1/2 percent of the main gas stream flow, respectively. For the two higher firing temperatures (1700 and 1920 K (2600° and 3000° F)), the compressor exit air used for cooling was precooled to 420 K (300° F).

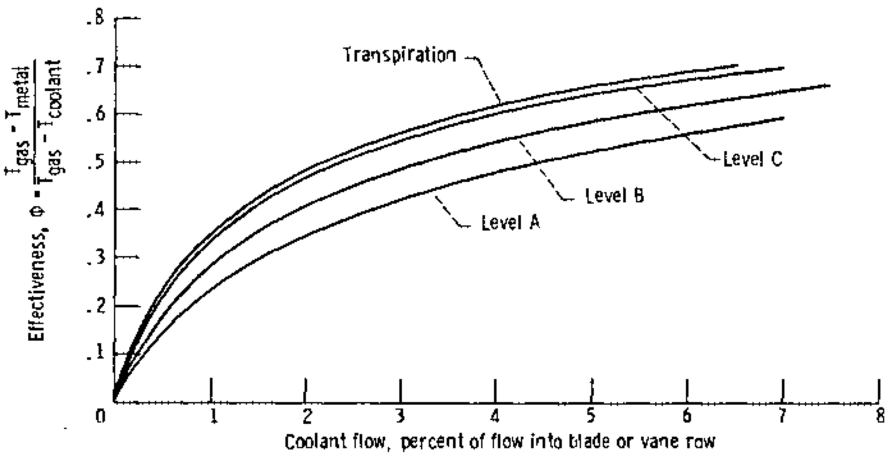

Figure 8. - Effectiveness as a function of coolant flow for four gas turbine cooling technologies.

APPENDIX C

SIMPLIFIED ANALYTICAL IGCC SYSTEM MODEL

The simplified energy flow model used to develop a system efficiency relationship for a coal-gasifier combined cycle is shown in figure 6.

SYMBOLS

a fraction of raw-gas chemical energy that enters gas turbine

b fraction of raw-gas chemical energy lost in cleanup system

f fraction of Q_{exh} transferred to steam cycle

K^i influence coefficient for p^i

p^i parameter i

P_{aux} auxiliary power requirements

P_{gt} gas turbine power

P_{net} net system power

P_{st} steam turbine power

Q_c coal energy input based on higher heating valve

Q_{ch} chemical energy in raw gas

Q_{exh} energy content of gas-turbine exhaust

Q_g energy input to steam cycle from gasifier

Q_{gt} fuel energy to gas turbine, $=aQ_{ch} + yQ_{th}$

Q_m energy input to steam cycle from the cooling of compressor discharge air used for turbine cooling

Q_{rg} energy content of raw gas $=Q_{ch} + Q_{th}$

Q_{st} steam-turbine energy, $=xQ_{th} + fQ_{exh} + Q_g + Q_m$

Q_{th} thermal energy in raw gas

x fraction of raw-gas thermal energy directly transferred to steam cycle

y fraction of raw-gas thermal energy directly transferred to gas turbine

z fraction of raw-gas thermal energy that does not directly enter either gas turbine or steam turbine

η_{cc} net IGCC system efficiency

η_{gt} net gas-turbine efficiency

η_{st} net steam-turbine efficiency

We may define net IGCC system efficiency as

$$\eta_{cc} = \frac{P_{net}}{Q_c} = \frac{P_{gt}}{Q_c} + \frac{P_{st}}{Q_c} - \frac{P_{aux}}{Q_c}$$

$$= \left(\frac{P_{gt}}{Q_{gt}} \frac{Q_{gt}}{Q_c}\right) + \frac{P_{st}}{Q_{st}} \frac{Q_{st}}{Q_c} - \frac{P_{aux}}{Q_c}$$

$$= \eta_{gt} \frac{Q_{gt}}{Q_c} + \eta_{st} \frac{Q_{st}}{Q_c} - \frac{P_{aux}}{Q_c}$$

where

$$Q_{gt} = aQ_{ch} + yQ_{th}$$

$$Q_{st} = xQ_{th} + fQ_{exh} + (Q_m + Q_{gt})$$

Substituting yields

$$\eta_{cc} = \eta_{gt} \left(a \frac{Q_{ch}}{Q_c} + y \frac{Q_{th}}{Q_c}\right) + \eta_{st} \left(x \frac{Q_{th}}{Q_c} + f \frac{Q_{exh}}{Q_c} + \frac{(Q_m + Q_g)}{Q_c}\right) - \frac{P_{aux}}{Q_c}$$

But

$$\frac{Q_{exh}}{Q_c} = \frac{Q_{exh}}{Q_{gt}} \frac{Q_{gt}}{Q_c} = \frac{Q_{exh}}{Q_{gt}} \left(a \frac{Q_{ch}}{Q_c} + y \frac{Q_{th}}{Q_c}\right)$$

so

$$\eta_{cc} = \left(\eta_{gt} + \eta_{st} \frac{Q_{exh}}{Q_{gt}}\right) a \frac{Q_{ch}}{Q_c}$$

$$+ \left[x\eta_{st} + y\left(\eta_{gt} + f\eta_{st} \frac{Q_{exh}}{Q_{gt}}\right)\right] \frac{Q_{th}}{Q_c} + \left(\frac{Q_m + Q_g}{Q_c}\right) \eta_{st} - \frac{P_{aux}}{Q_c} \qquad \text{(C1)}$$

General Influence Coefficients

Equation (C1) indicates that the net IGCC system efficiency is a function of 11 parameters:

$$\eta_{cc} = \eta_{cc} \left(a, x, y, z, f, \frac{Q_{ch}}{Q_c}, \frac{Q_{th}}{Q_c}, \eta_{gt}, \eta_{st}, \frac{Q_m + Q_g}{Q_c}, \frac{P_{aux}}{Q_c}\right)$$

A general influence coefficient can be defined as being the relative change in net system efficiency caused by a relative change in one of the 11 parameters; or

$$\frac{\partial \eta_{cc}}{\eta_{cc}} = K^i \frac{\partial p^i}{p^i}$$

where K^i, the influence coefficient for p^i, is defined as

$$K^i \equiv \frac{\partial \eta_{cc}}{\partial p^i} \frac{p^i}{\eta_{cc}}$$

Coefficient K^i can be evaluated from equation (C1) by differentiating with respect to one of the 11 parameters and holding the remaining 10 parameters constant. In order to obtain the influence coefficient for η_{gt} and x, equation (C1) will be modified by assuming that $Q_{exh}/Q_{gt} = 1 - (\eta_{gt} + C_1)$, where C_1 is a constant that includes gas-turbine combustor, generator, and mechanical losses, and by making use of the relation $x + y + z = 1$ and assuming that z = constant.

In an IGCC system using a particular gasifier a, Q_{ch}/Q_c and Q_{th}/Q_c would be constant parameters. The more significant influence coefficients for the IGCC systems considered in this analysis are

$$K_{gt} = \frac{\partial \eta_{cc}/\eta_{cc}}{\partial \eta_{gt}/\eta_{gt}} = \left(1 - f\eta_{st}\right)\left(a \frac{Q_{ch}}{Q_c} + y \frac{Q_{th}}{Q_c}\right)\frac{\eta_{gt}}{\eta_{cc}} \tag{C2}$$

$$K_{st} = \frac{\partial \eta_{cc}/\eta_{cc}}{\partial \eta_{st}/\eta_{st}} = \left[\left(a \frac{Q_{ch}}{Q_c} + y \frac{Q_{th}}{Q_c}\right) f \frac{Q_{exh}}{Q_{gt}} + x \frac{Q_{th}}{Q_c} + \frac{Q_m + Q_g}{Q_c}\right]\frac{\eta_{st}}{\eta_{cc}} \tag{C3}$$

$$K_y = \frac{\partial \eta_{cc}/\eta_{cc}}{\partial y/y} = \left[\eta_{gt} - \eta_{st}\left(1 - f \frac{Q_{exh}}{Q_{gt}}\right)\right]\frac{Q_{th}}{Q_c}\frac{y}{\eta_{cc}} \tag{C4}$$

$$K_f = \frac{\partial \eta_{cc}/\eta_{cc}}{\partial f/f} = \eta_{st} \frac{Q_{exh}}{Q_{gt}}\left(a \frac{Q_{ch}}{Q_c} + y \frac{Q_{th}}{Q_c}\right)\frac{f}{\eta_{cc}} \tag{C5}$$

$$K_{(Q_m+Q_g)/Q_c} = \frac{\partial \eta_{cc}/\eta_{cc}}{\dfrac{\partial \left[(Q_m + Q_g)/Q_c\right]}{(Q_m + Q_c)/Q_c}} = \frac{\eta_{st}\left[(Q_{ch} + Q_g)/Q_c\right]}{\eta_{cc}} \tag{C6}$$

$$K_{P_{aux}/Q_c} = \frac{\partial \eta_{cc}/\eta_{cc}}{\frac{\partial (P_{aux}/Q_c)}{P_{aux}/Q_c}} = -\frac{P_{aux}/Q_c}{\eta_{cc}} \tag{C7}$$

APPENDIX D

ENERGY FLOW DIAGRAMS AND ENERGY FLOW RATIOS

FOR CASES COMPARING GASIFIERS

TABLE X. - ENERGY RATIO COMPARISONS

	Case designation				
	WA17CB	TA17CB	WP17CB	TP17CB	BO17CB
Gasifier/cleanup subsystems:					
Raw fuel gas output (total)/coal input	1.0643	1.0686	1.2559	0.9924	1.0042
Raw fuel gas (HHV)/coal input	0.8418	0.6666	1.0817	0.7893	0.9587
Raw fuel gas (HHV)/raw fuel (total)	0.7910	0.6238	0.8613	0.7953	0.9547
Raw fuel gas (sensible)/raw fuel (total)	0.1965	0.3358	0.1191	0.1740	0.0392
Raw fuel gas (latent)/raw fuel (total)	0.0126	0.0403	0.0196	0.0307	0.0061
Clean fuel to combustor (total)/coal input	0.8077	0.7099	0.8258	0.7971	0.9555
Clean fuel (HHV)/coal input	0.7562	0.6467	0.7995	0.7695	0.9298
Clean fuel (sensible)/coal input	0.0495	0.0624	0.0253	0.0274	0.0252
Clean fuel (latent)/coal input	0.0020	0.0008	0.0010	0.0002	0.0005
Gasifier oxident input/coal input	0.0361	0.0839	0.0064	0.0034	0.0011
Gasifier steam/water input/coal input	0.0243	-0.0008	0.0486	-0.0008	0.0310
Recycled gas/coal input	0.0590	0	0.246	0	0
Gasifier auxiliary power/coal input	0.0046	0.0052	0.0046	0.0052	0.0020
Gasifier oxidant delivery system:					
Gas turbine compressor extraction air/ oxidant input	1.2946	0.8216	------	-------	-------
Heat added to oxidant/oxidant input	0	0.2322	------	-------	-------
Boost compressor power/oxidant input or oxygen plant auxiliary power/oxidant input	0.2965	0.1881	4.2319	11.9091	19.1250
Gas turbine subsystem:					
Gas turbine generator output/fuel input (total)	0.2934	0.2792	0.3149	0.3182	0.3169
Generator output/fuel input (HHV)	0.3133	0.3065	0.3252	0.3297	0.3256
Extraction air for gasifier/fuel input (total)	0.0579	0.0970	------	-------	-------
Gas turbine exhaust (total)/fuel input (total)	0.5850	0.5675	0.6136	0.6104	0.6118
Gas turbine exhaust (sensible)/fuel input (total)	0.5248	0.5123	0.5437	0.5494	0.5490
Gas turbine exhaust (latent)/fuel input (total)	0.0602	0.0553	0.0699	0.0610	0.0629
Steam turbine subsystem:					
Steam turbine generator output/net heat input	0.4158	0.4126	0.3991	0.4148	0.384
Net heat input from HRSG/net heat input	0.6910	0.4736	0.7911	0.6754	0.887
Net heat input from RGC/net heat input	0.2641	0.4854	0.1632	0.2679	-------
Net heat input from miscellaneous/net heat input	0.0449	0.0410	0.0458	0.0567	0.113
Steam generators:					
Raw fuel gas cooler (RGC):					
Net RGC heat to steam cycle/coal input	0.1310	0.2824	0.0749	0.1375	-------
Economizer heat/net RGC heat to steam	0	0.2396	0	0.0705	-------
Boiling heat/net RGC heat to steam	0.3315	0.3092	0.3316	0.5603	-------
Superheating/net RGC heat to steam	0.6685	0.4512	0.6684	0.3692	-------
Fuel gas recuperation/coal input	0.0314	0.0394	0.0162	0.0174	[a]0.0233
Net heat to gasifier steam/coal input	0.0177	------	0.0387	------	[a]0.027
Heat recovery steam generator (HRSG):					
Net HRSG heat to steam cycle/coal input	0.3426	0.2756	0.3630	0.3466	0.372
Economizer heat/net HRSG heat to steam	0.5232	0.5318	0.4151	0.4942	0.376
Boiling heat/net HRSG heat to steam	0.0917	0	0.1099	0	0.138
Superheat/net HRSG heat to steam	0.1848	0.1766	0.2215	0.3018	0.278
Reheat/net HRSG heat to steam	0.2000	0.2916	0.1626	0.2040	0.140
Induction heating/net HRSG heat to steam	0	0	0.0916	0	0.069
Heat added to oxidant/coal input	0	0.0195	0	0	0
Overall system:					
Gross generator output/coal input	0.4431	0.4383	0.4431	0.4665	0.464
Auxiliary power requirement/coal input	0.0355	0.0425	0.0506	0.0665	0.043
Net electrical output/coal input	0.4077	0.3957	0.3926	0.4000	0.421

[a]In HRSG.

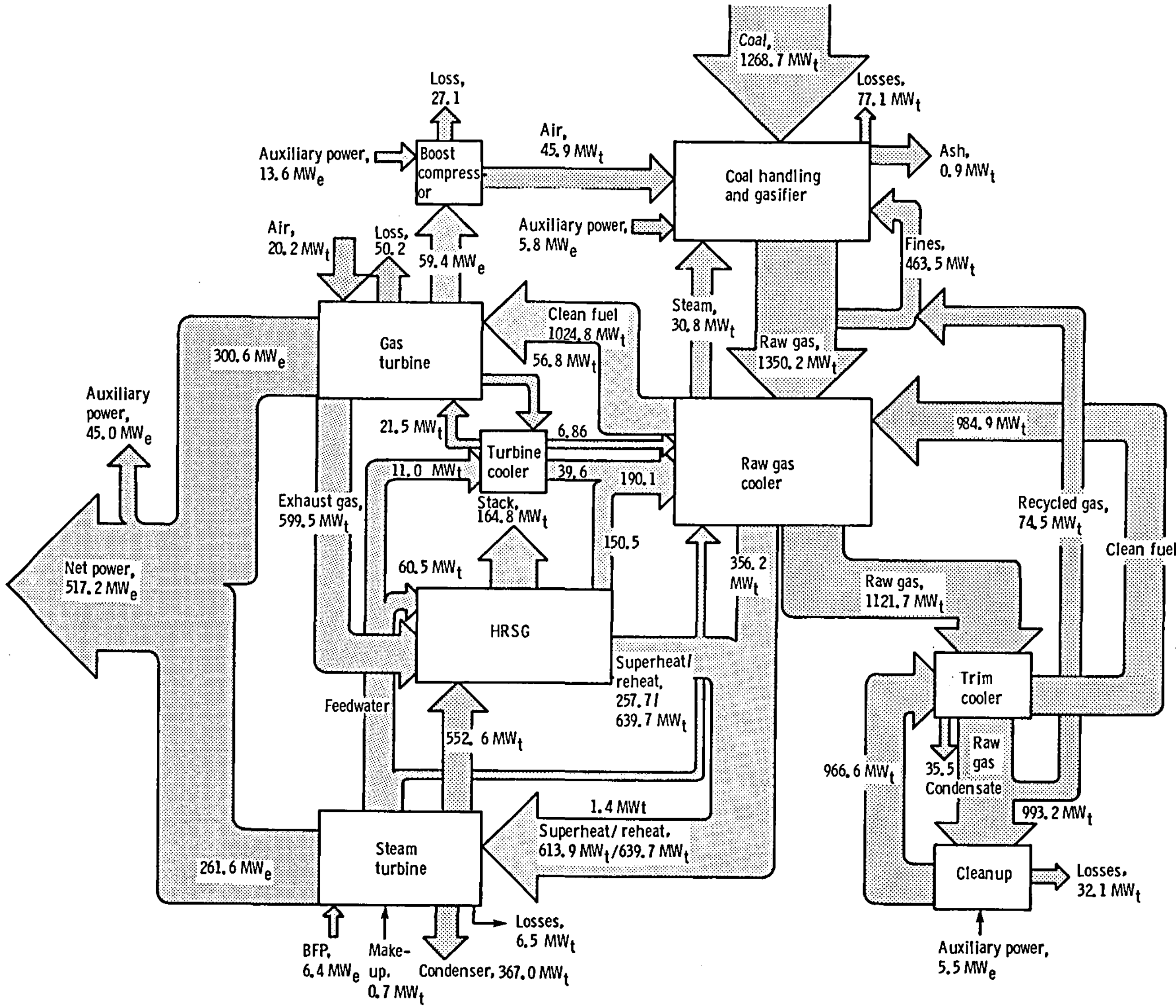

Figure 9. - Energy flow diagram for case WA17CB.

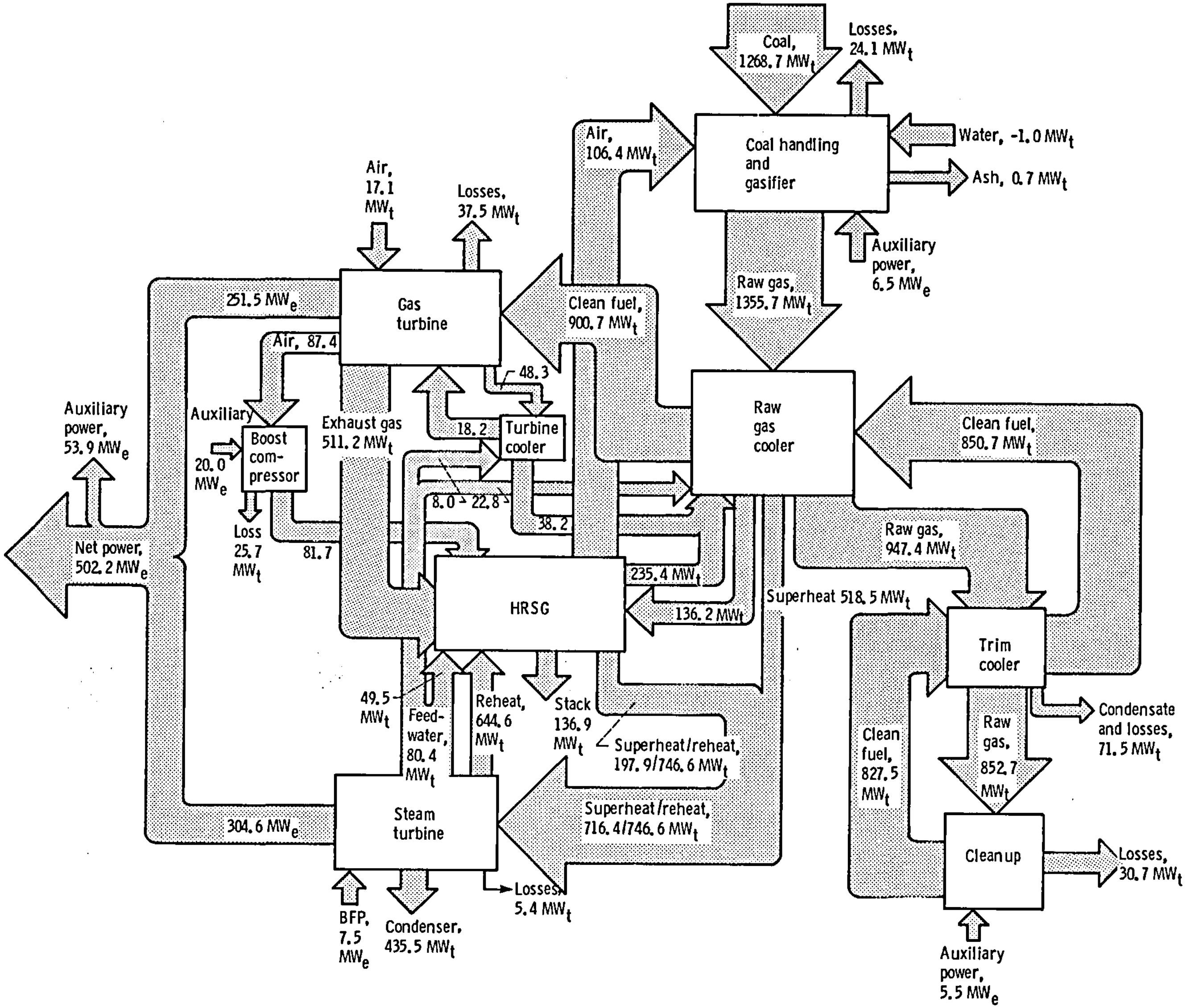

Figure 10. - Energy flow diagram for case TA17CB.

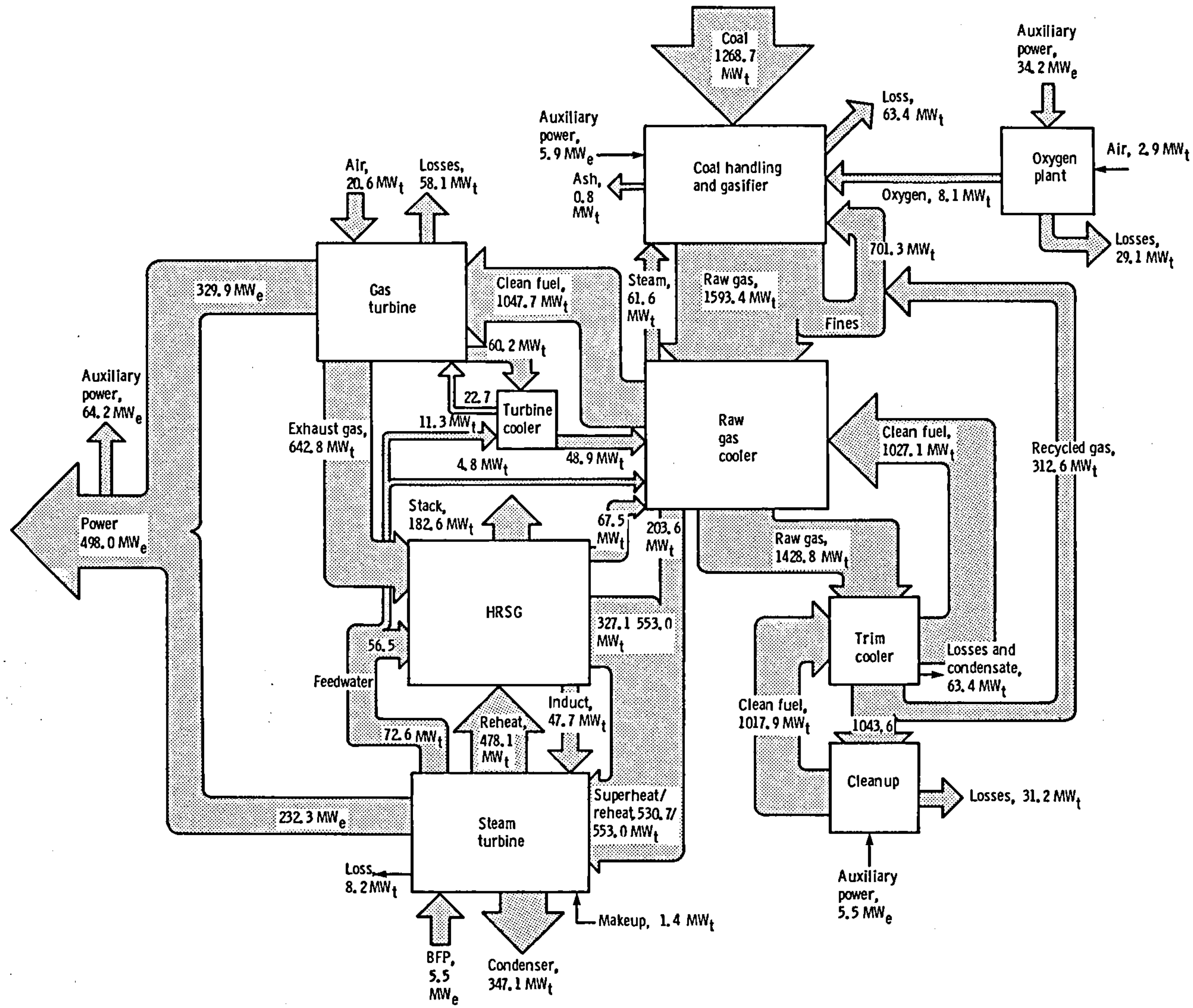

Figure 11. - Energy flow diagram for case W017CB.

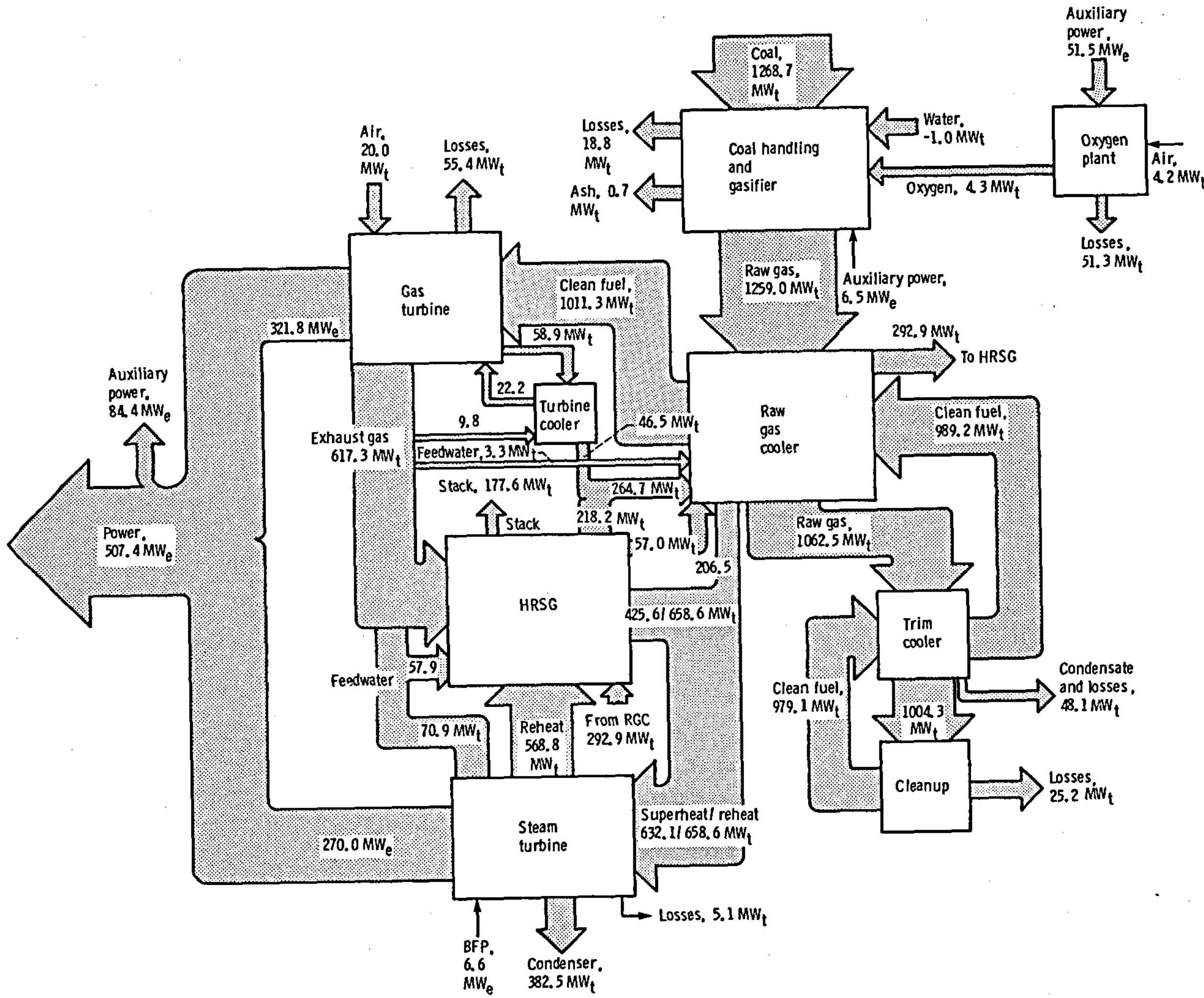

Figure 12. - Energy flow diagram for case TO 17CB.

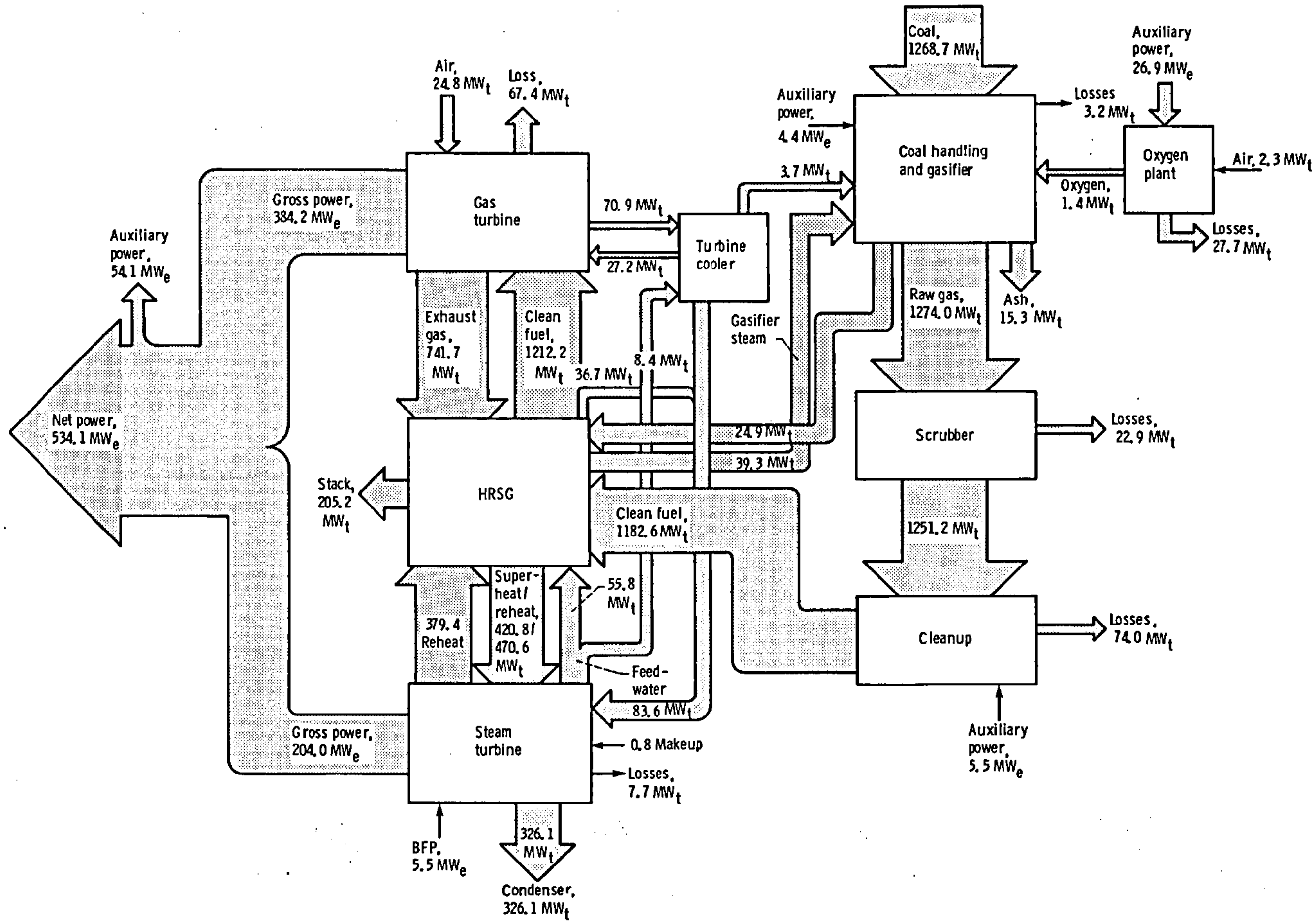

Figure 13. - Energy flow diagram for case BO 17CB.

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16. Abstract Performance estimates were made for a series of integrated coal gasification combined-cycle (IGCC) power systems using three generic types of coal gasification subsystems. The objectives of this study were (1) to provide a self-consistent comparison of IGCC systems using different types of gasifiers and different oxidants and (2) to use this framework of cases to evaluate the effect of a gas-turbine firing temperature and cooling approach an overall system efficiency. The basic IGCC systems considered included both air- and oxygen-blown versions of a fluidized bed gasifier, represented by the Westinghouse design, and an entrained-bed gasifier, represented by the Texaco design. Also considered were systems using an oxygen-blown, fixed-bed gasifier, represented by the British Gas Corporation (BGC) slagging gasifier. All of these gasifiers were integrated with a combined cycle using a gas-turbine firing temperature of 1700 K (2600° F) and a compressor pressure ratio of 16:1. Steam-turbine throttle conditions were chosen to be 16.6 MPa/811 K (2400 psia/1000° F) with a single reheat to 810 K (1000° F). Some of these cases were modified to allow the evaluation of the effect of gas-turbine firing temperature. Turbine firing temperatures from state-of-the-art 1365 K (2000° F) to an advanced-technology 1920 K (3000° F) were analyzed. A turbine-cooling technology that maintains metal temperatures below acceptable limits was assumed for each level of firing temperature. System performance comparisons were made using three advanced turbine-cooling technologies for the 1920 K (3000° F) firing temperature. The results indicate that the IGCC using the BGC gasifier had the highest net system efficiency (42.1 percent) of the five gasification cases considered. The other four cases had net system efficiencies in the 40.0-percent range. In all cases, net system efficiency increased as firing temperatures increased. The increase is greatest for the fixed-bed gasifier because of the higher chemical energy content of its fuel gas. The air-blown version of the fluid-bed gasifier has an efficiency advantage over its oxygen-blown counterpart, but the advantage decreases as the firing temperature increases. The use of advanced convective cooling technology, at a 1920 K (3000° F) firing temperature, results in higher system performance than either water cooling in the fixed-bed system or transpiration cooling in the fluid-bed gasifier system. Sensitivity parameters developed from the cases using the 1700 K (2600° F) gas-turbine firing temperature indicate that it is particularly important to improve the individual performance of both the gas turbine and steam turbine subsystems. The gas-turbine system performance can be improved by increasing firing temperature in conjunction with advanced cooling techniques. The steam-turbine system can be improved by careful selection of throttle and reheat conditions that can use the high-quality heat available from the raw gas cooler and the heat-recovery steam generator.

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